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Indian Standard

METHODS FOR MEASUREMENT OF PETROLEUM GASES AND TOWN GAS IN BULK

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METHODS FOR MEASUREMENT OF PETROLEUM GASES AND TOWN GAS IN BULK

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Indian Standard
METHODS FOR MEASUREMENT OF PETROLEUM GASES AND TOWN GAS IN BULK

0. FOREWORD

0.1 This Indian Standard was adopted by the Indian Standards Institution on 30 April 1977, after the draft finalized by the Petroleum Measurements Sectional Committee had been approved by the Chemical Division Council.

0.2 Accurate measurement of bulk quantities of petroleum gases and town gas is essential for accounting of stock supplied or received, loss control, customs, excise and other similar purposes. Bulk transfer of gas may be for the purposes of supplying fuel, feed stock or component sales, for example, C_2H_4, C_3H_6, C_4H_8, etc. The pricing clause in a contract between the supplier and the purchaser lays down the basis for billing, and this generally depends on the end use of the gas supplied. For instance, if the end use is fuel, then the calorific value delivered is of primary concern. If the end use is feed stock, then the volume or mass is of prime importance. If the end use is extraction of a particular component, then composition together with the volume delivered acquires precedence over other considerations. Hence, dependent upon the pricing clause and end use of the gas, the methods of measurement will vary. The directions given in this standard lay down procedures, methods and equipments to be used for determining the quantities of these gases for various uses. While no single set of standard methods can cover all conditions and requirements which may arise in practice, the various methods and procedures described herein are intended to provide a general standard of uniformity. It should be noted that petroleum and town gases are highly explosive and every effort shall be made to ensure safety of personnel and plant.

0.3 For ease of reference, all tables are given at the text of the standard.

0.4 Taking into consideration the views of interests concerned, the Sectional Committee responsible for the preparation of this standard felt that it should be related to the trade practices followed in the country in this field. Furthermore, due weightage has to be given to the need for international coordination among standards prevailing in different countries of the world in this field. This standard has, therefore, been developed utilizing mainly the recommendations of International Organization for Standardization (ISO), the standards and publications of the American Gas Association (AGA), and the American Society of Mechanical Engineers (ASME). In the preparation of this standard, assistance has been derived from the following publications, which is gratefully acknowledged:

ISO/R 541 Measurement of fluid flow by means of orifice plates and nozzles. International Organization for Standardization.

ISO/R 781 Measurement of fluid flow by means of venturi tubes. International Organization for Standardization.

JACKSON (R M), JOHNSON (R R), LARSON (C A) and STEARNS (R F). Flow measurement with orifice meters. *The Esso Series.* D. Van. Nostrand Company, Inc, New York.

RHODES (T J). Industrial instruments for measurement and control. *Chemical Engineering Series.* McGraw-Hill Book Company, Inc, New York.

Gas Committee Report No. 3. Orifice metering of natural gas. American Gas Association.

Gas analysis and testing of gaseous material. American Gas Association.

Fluid meters — their theory and application. 1959. Report of ASME Research Committee on fluid Meters. American Society of Mechanical Engineers.

DOUGLAS (M. Considine). Process instruments and control handbook. McGraw-Hill Book Company, Inc, New York.

PERRY (R H). Chemical Engineers Handbook. 3 Ed. McGraw-Hill Book Company, Inc, New York.

Shell flow meter engineering handbook. Royal/Shell Group. Ed 1. 1968. Bataafse Internationale Petroleum Maatschappij N. V. Manufacturing, MFE.

0.5 In reporting the result of a test or analysis made in accordance with this standard, if the final value, observed or calculated, is to be rounded off, it shall be done in accordance with IS : 2-1960*.

1. SCOPE

1.1 This standard lays down methods for measurement of bulk quantities of petroleum gases and town gas. It prescribes:

a) the suitability of various available measuring devices for different conditions;
b) requirements for measuring devices for the measurement of petroleum gases and town gas, when stored and handled in bulk;
c) calculation;
d) sampling, wherever required for laboratory determination; and
e) calibration.

1.1.1 This standard excludes measurement of bottled gas and gaseous storage.

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*Rules for rounding off numerical values (*revised*).

2. INTRODUCTION

2.1 Different types of fluid meters, each with its own limitations, are available for measuring gas flow. Though these meters may differ to a great extent in their design and principle of operation, all of them consist of two distinct parts with their own distinct functions to perform. The first is the primary element, which is in contact with the fluid, resulting in some form of interaction. This interaction may be that of imparting motion to the primary element; the fluid may be accelerated; or there may be an exchange of heat. In doing so, the primary element develops a signal for the second or secondary element, which, in turn, analyses the signal and thus translates the interaction between fluid and primary element into volumes, masses, rates of flow, and indicates or records the result.

2.2 For example, in an orifice meter, the orifice, together with the adjacent part of the pipe and the pressure connections, constitute the primary element, while the secondary element consists of a differential pressure gauge together with some sort of mechanism for translating the pressure difference (which is the signal in this case from the primary element) into a rate of flow indicating the result, in some cases also recording graphically and integrating with respect to time. The same sort of combination will be observed for other types of meters.

2.3 The secondary devices may obviously be varied almost indefinitely, but the primary elements depend for their operation on a few simple physical principles. Therefore, fluid meters may best be classified with regard solely to the nature of the primary element or to the physical principles involved.

3. CLASSIFICATION

3.0 Flow meters for measuring gas flow may be classified as : (a) inferential types, and (b) volumetric types.

3.1 The inferential type of meter may be defined as a meter that obtains a measurement of the flow of gas, not by measuring the volume or mass of the medium but, by measuring some phenomenon, that is, a function of the quantity of gas passing through the pipe. The phenomenon usually measured is either pressure differential or velocity in the pipe.

3.1.1 Differential pressure flow meters are the largest and the most important group of flow meters, these are used to indicate the quantity of gas flowing by utilizing the difference in pressure resulting from a constriction in the pipe through which the gas is flowing. Differential pressure type of flow meters can be used for maximum flow pressure of 400 kg/cm^2 and a maximum flow temperature of 500°C. Different types of orifice meters, flow nozzles, venturi meters and pitot tubes are included in this group.

3.1.1.1 *Orifice meter* — This is by far the most widely used type of differential pressure meters, because these are relatively inexpensive, simple and flexible compared to the other types of flow meters. Orifice meters can be constructed from all metals and also from certain plastics. Orifice meters are easy to install, and a properly calibrated orifice meter can give an accuracy of ±0·75 percent, whereas accuracy for uncalibrated orifices may be ± 1·5 percent. (Table 3 gives the limitations of pipe size, Reynolds number, diameter ratio, etc, for different types of orifice meters.) However, the orifice meter has a greater pressure drop compared to the other types of flow meters and it gives poor accuracy for high ratios of orifice size to pipe size. The primary element for an orifice meter is the orifice plate. Though various types of orifice plates such as quarter circle orifice, conical entrance orifice, segmental orifice, etc, are available for various specialised use, the thin plate square edge type is still the most common, and this standard will only consider the thin plate square edge orifice within its scope. A standard orifice plate is characterised by the arrangement of pressure taps (*see* **9.2**, **9.5** and **9.6**). No orifice plate shall be used according to this standard unless it conforms with the description given in **9** of this standard.

3.1.1.2 *Venturi meters* — Venturi meters have the specific advantage of low pressure loss and greater accuracy for flow measurement. High initial cost, lesser flexibility and the longer length of primary element are some of the specific disadvantages of a venturi meter. Venturi meter with the classical venturi tube as primary element is the most common. In certain cases rectangular venturi tubes and dall tubes are also used as the primary element, but, those will not be considered within the scope of this standard. The range of use for a classical venturi tube within the compass of this standard depends on the way they are manufactured as indicated in **11.1.1, 11.1.2** and **11.1.3.**

3.1.1.3 *Nozzles* — Flow nozzle is an adoption of the venturi tube that is often used to measure flow by means of differential pressure measurements. The pressure recovery for a nozzle is considerably less than that for the venturi tube but greater than that of the orifice plate. It has accuracy with high ratios of orifice to pipe size, and can be used effectively where erosion or cavitation prohibits the use of square edge orifices. ISA 1932 nozzle, long radius nozzle and venturi nozzle are the three common types of standard nozzles which differ considerably in their design, and are described separately in **10.1, 10.2** and **12.**

3.1.1.4 *Pitot tube* — It is a tube with an open end placed in the direction of the gas stream. The difference between the pressure in the pitot tube (impact pressure) and the static pressure in the line is a measure of the rate of flow. The pitot tube has very low pressure loss. A commercial type of pitot tube may be inserted or withdrawn while the line is in operation. Pitot tube finds application specially for the measurement of flow where the size of the pipe or duct is too great to make an orifice installation practical, and also for measurement of flow where a temporary installation is satisfactory. The major objection to the pitot tube is that any foreign matter, such as dirt or solids in the gas, will either clog the tube completely or build up on the exposed surface to such an extent as to seriously change the calibration of the instrument. As such, the pitot tube has a very poor accuracy, and is therefore not recommended for measuring town gases, or petroleum gases in bulk. Therefore, any further detail of this type of flow measuring device will not be discussed in this standard.

3.1.2 *Area Type Flow Meters* — A second general type of flow meter which may be described as inferential, and which is closely associated in theory with the orifice type of meter, is the area type gas flow meter. The fundamental difference between the orifice and the area type of gas meter is the fact that the orifice type infers the quantity of flow from the pressure differential across an orifice or constriction in the pipeline of constant dimensions, whereas the area type of flow meter infers the quantity of flow from changes in the area of a constriction or orifice in the line across which the pressure differential is constant. Rotameter and piston type area meters are the two common types in use. In general, area type flow meters are moderately expensive and get damaged by fluid shocks.

3.1.2.1 *Rotameter* — It consists essentially of a plummet or float which is free to move in a vertical, slightly tapered tube with the small end down. The fluid enters the lower end of the tube and causes the plummet to rise until the annular area between the plummet and the wall of the tube is such that the pressure drop across this constriction is just sufficient to support the plummet. Typically, the tapered tube is of glass and carries etched upon it a linear scale on which the position of the floating plummet indicates the flow. Rotameters are available with pneumatic, electric, and electronic transmitters for actuating remote recorders, integrators and automatic flow controllers.

Interchangeable precision-bore glass tubes and metal metering tubes are available. Rotameters have proved satisfactory up to a maximum pressure of 400 kgf/cm^2 absolute and a maximum temperature of 250°C. A single rotameter can readily cover a ten-fold range of flow, and by providing more than one float a two hundred fold range is practicable. Rotameters require no straight runs of pipe before or after the point of installation. Pressure loss remains constant over the whole flow range.

In experimental work, for the greatest precision, a rotameter should be calibrated with the fluid which is to be metered. However, modern rotameters not only have predictable calibrations but are immune to effect of viscosity variations over a wide range of Reynolds numbers, so that viscosity variations usually cause no alteration of the calibration. Rotameters can also be rendered immune to fluid density changes of as much as 15 percent. For either manual or automatic control, rotameter permits visualization of the flow process by the operator and comparison between primary and secondary instrument without shutting down. Rotameters are expensive in the large sizes and are recommended. to be used between 6 to 300 mm line size. Accuracy of an uncalibrated rotameter is about ±2 percent of the full scale value, whereas ±1 percent accuracy can be obtained from properly calibrated rotameter.

Great care shall be used in installing the rotameter tube in order to prevent leakage at the packing joints, and to prevent mechanical stress which may deform or break it. Care should also be taken to install the meter where it can be firmly fastened to a bracket or panel in order to prevent stresses due to piping distortion which might cause damage to the tube. It is desirable that rotameters be installed within 3 or 4 degrees of a vertical position in order to avoid reduction in the axial component of the float weight.

3.1.2.2 *Piston type area meter* — The principle of this meter is very similar to that of a rotameter, although its general appearance is entirely different. In this instrument the fluid enters the meter-body horizontally, is deflected against a floating piston, and passes through an orifice on the side opposite the inlet. The orifice is rectangular in shape and fixed on the bottom and sides. The top of the orifice is formed by the piston, and as the latter moves up or down, the size of the orifice is varied accordingly. Similar to the rotameter plummet, the movement of the piston in this type of meter indicates the flow rate.

Proper installation of a piston-type area meter requires that there be at least 20 pipe diameters of straight pipe ahead of the meter. It will be noted that this requirement is not needed in case of the rotameter.

3.1.3 *Anemometers* — This type of instrument consists of a set of rotating vanes placed at an angle of about 45° to the axis of flow and free to rotate about an axis set in jewelled bearings. The rotating shaft in turn operates a counting mechanism which registers the number of revolutions of the vanes. With a knowledge of the line size, pressure, temperature and the calibration of the anemometer, the number of revolutions for a certain definite period can be translated to the quantity of gas flow over that specific period. If an anemometer is set up in a pipe line, the distribution of velocity will be upset by the instrument; and since the effective area of the pipe is reduced by the instrument, the velocity will be higher at this point and the anemometer will give a false reading. In this case the meter has to be calibrated in the pipe if any degree of accuracy is to be attained. If the pipe is large with respect to the instrument, say 91 cm in diameter or better, then the error resulting from a change in velocity at the meter will be negligible. The objection to the anemometer is that it is not accurate. This is due to the fact that the friction in the mechanism depends on the condition of the instrument and any lack of oil or any dirt in the gearing will seriously affect the calibration.

Furthermore, the instrument is completely reliable only over the range in which it has been calibrated, since the friction forces will vary with the speed of rotation. The magnitude of inaccuracies that may develop in measuring flows with an anemometer may be as great as 10 to 20 percent, depending on the accuracy of calibration of the instrument and the character of the flow measured. This meter is, therefore, not recommended for measurement of town gas and petroleum gas in bulk and is excluded from the scope of this standard.

3.1.4 *Electrical Air-Conductance Meters* — By utilising the ability of the gas to conduct heat from a wire or grid heated electrically it is possible to obtain a quantitative measurement of gas flowing through a pipe line. Since the ability of the gas to conduct heat will vary with the velocity, this fact can be used to determine the rate and consequently the quantity of flow through the pipe. Hot wire anemometer and the Thomas meter are the two types of instrument which utilise, electrical conductance as a measure of flow. However, in general this is not so practical a method of determining gas flow as other methods previously described, and is not widely used. This, therefore, is excluded from the scope of this standard.

3.2 Volumetric Type of Fluid Meters — Volumetric meter measures the flow rate by separating the fluid into individual portions by mechanical means and counting these segments. These meters are also known as positive displacement meters or simple displacement meters. This group includes a rather large number of instruments with diverse mechanical features and it is beyond the scope of this standard to discuss them in detail. Out of a wide variety of meters belonging to this group, the geared or lobed impeller (rotary) type, the bellows type, the roots type and the liquid sealed drum type are more common. A brief discussion of these types is covered in **3.2.1** to **3.2.4**.

The volumetric meter is accurate on small rates of flow and is not seriously affected by changes in pressure, density or viscosity. These meters have a simple integrator and the installation cost is low. They are compact, portable and protected against possible abuse or accidental breakage. Meters of this group are expensive specially in large sizes. They lack the flexibility obtained with the orifice meter which can be changed easily from large to small pipe line. They are expensive to repair. The meters belonging to this group are all mechanical devices. Therefore, a high degree of accuracy for the manufacture of a volumetric or a current type of meter is important. To ensure this, the meter should be certified by recognised authorities such as National Instruments, or should be procured from an ISI approved manufacturer.

3.2.1 *Geared or Lobed Impeller Type Meter* — The geared or lobed impeller (rotary) type of meters usually consist of two rotors, each with two or more lobes, mounted so as to rotate in a manner similar to gears with almost rolling contact. The inner surface of the case and the impeller tips are carefully machined to reduce clearance to the minimum.

3.2.2 *The Bellows Type of Meter* — The bellow type of volumetric flow meter is designed primarily and exclusively for gas measurement and is by far the most widely used volumetric meter for the service. These meters have two or more movable partitions, called diaphragms, attached to the case by a flexible material (usually leather) so that each partition may have a reciprocating motion. This motion of the diaphragm not only operates the secondary element but also the valves controlling the flow of gas in and out of the meter. Adjustment is effected by regulating the magnitude of the movement of the diaphragms.

The size and design of the valves are of considerable importance in the bellows type of meter. If the meter is used on a gas that carries considerable tar or sticking material, it is necessary to build it with small valves to ensure sufficient power to prevent sticking. If the gas is pure and free from foreign material the valve should be large enough to ensure maximum capacity for a meter of given bellow size. The pressure differential across a bellows type of meter is always small, and there is little or no change in design of the working parts for meters of different working pressures. However, if the meter is to operate on a high pressure system, it is necessary to design the outer meter case to withstand safely the maximum working pressure.

The same problem exists in the case of the volumetric gas meter as was found in the orifice meter, of computing the flow of gas in terms of a standard basic temperature and pressure. It is evident that unless the pressure of the gas is known, the accuracy of the meter will be in doubt. Since pressure variations are likely to occur in a gas system supplying a chemical plant or a chemical process, it is important that means be taken to determine and record these variations whenever the meter does not have a mechanism to correct automatically the readings for pressure variations. The accuracy of the bellows type of gas meter is a variable quantity depending on the accuracy with which it is calibrated, the conditions of the slide valves, the tightness of the bellows diaphragms, the extent of pressure variations and the excellence of the design and construction of the meter. In general, the meter should be accurate within 1 percent over its range when properly installed and in perfect repair. If the valves are worn, the bellows cracked, or the mechanism sticking, the error may increase considerably. An average accuracy over the life of the meter should be about 2 percent over its range. Gas meters of this type require periodic attention if they are to be kept within tolerance at all times.

3.2.3 *The Liquid-Sealed Rotary Gas Meter* — This consists of a drum shaped meter body slightly more than half full of liquid (usually water). A rotor with spirally shaped vanes is mounted in the centre of the body. The gas enters the meter body through the centre of the rotor shaft and below the level of the liquid which is maintained somewhat above the shaft. It then discharges through a short vertical pipe just above the liquid level after which it is trapped in the chamber formed by the spiral vane which has both ends submerged under the liquid. The pressure of the gas causes the rotor to revolve. When one vane emerges from the liquid and releases the pressure, the next vane forms a closed chamber and continues to cause the rotor to revolve. If the liquid level is exactly correct, there will be no by-pass, and the rotation of the rotor will give a true indication of the volume of gas discharged. This meter is particularly accurate on small rates of flow and can be successfully used up to a maximum rotative speed of 100 rev/min. When accurately calibrated and with proper liquid level, it will develop a probable error of approximately 0·5 percent only. If the gas flow is fluctuating, it may produce in the meter a surging of the liquid which will cause serious error; and if this condition exists it is necessary to provide a gas bag or surge tank to smooth out the flow. The inherent difficulty with this meter is the fact that the liquid level as well as the meter level is highly critical. Therefore, any condensation of the gas, leakage of the liquid from the meter or any evaporation due to the measuring of a dry gas will quickly destroy the calibration. For this reason, in spite of its accuracy, the meter is not widely used in commercial or process metering and hence, excluded from the scope of this standard.

3.2.4 *The Roots Type of Meter* — It consists of a set of two rotors having a cross-sectional area (at right angles to the rotating shaft) in the approximate shape of the figure eight. The rotors are so mounted in the meter body as to mesh at right angles to each other by means of two gears mounted outside the meter body on extension of the rotor shafts. The body is shaped in such a way as to provide a close fit between it and the rotor edge at all points. The gas is admitted at the top of the meter, and the head pressure causes the rotor to revolve. In so doing they trap a certain amount of gas between the rotors and the meter body. The gas is prevented from by-passing the meter between the rotors by the close mesh of the rotors, which, almost but not quite, touch at all times. As a result, this meter is a true volumetric meter, and the revolutions of either shaft are a direct indication of the flow.

However, since the parts of the meter do not touch at any point, there is always a small amount of gas by-passing it, and this has to be accounted for in the calibration. Since this leakage will vary as the square root of the pressure differential, it will not be constant over the entire range of the meter and cannot be completely compensated for by means of a change in the counter gear ratio. Special attention shall be paid to providing friction free bearings and light rotors for greater accuracy of the meter. If the meter is in good mechanical condition and has close clearance, the slippage error will not throw it out of tolerance over a range of at least 95 percent of maximum capacity. On rates of flow of less than 5 percent of full capacity, it is necessary to provide a correction factor to account for the slippage of gas past the rotors. The condition of the meter can be checked by blanking off its outlet and rotating the meter just fast enough to maintain the pressure differential existing under operating conditions. The number of rev/min required to hold this differential will be a measure of the volume of air by-passing the meter, and will indicate the correction necessary to obtain the true flow. This may easily be computed from the known volumetric capacity of the meter. Periodic tests for slippage will indicate the increased error resulting from wear of the rotors due to dirt or abrasive material passing through the meter.

This type of gas meter is more common than the liquid sealed type and is usually used for measuring very large rates of flow. For calibration, volumetric measurement of the meter discharge by means of a gas holder is only used when a very high accuracy is required, otherwise the meter is usually corrected on the basis of the increased number of rpm required to maintain the operating pressure differential against a sealed discharge line above the number required when the meter was new.

4. UNITS AND REFERENCE CONDITIONS

4.1 Units of Gas Measurement — The unit of measurement shall be either cubic metre or kilogram or kilocalories, depending upon the end use of the gas and on the agreement between the buyer and the seller. However, a cubic metre of gas has no absolute or comparative value unless the pressure and temperature of the gas are specified.

4.2 The reference or standard or base temperature used in the measurement of commercial gases shall be 15°C.

4.3 The reference pressure to be used shall be 760 mm of mercury absolute at 0°C. The reference conditions with respect to the amount of water vapour with the gas shall be either complete water saturation or dry depending upon the agreement between the parties concerned.

5. MEASUREMENT OF FLUID FLOW BY MEANS OF ORIFICE PLATES, VENTURI TUBES AND NOZZLES

5.0 It is necessary to ensure that all the following requirements are completely fulfilled during the period of measurement. In this standard, a nozzle shall, if not otherwise specified, mean either ISA 1932 or long radius nozzle.

5.1 Primary Element

5.1.1 The primary element shall be manufactured, installed and used in accordance with this standard.

5.1.2 The condition of the primary element shall be checked after reasonable intervals. Also, refer to **15** for calibration and testing. Attention is drawn to the fact that gases, even when apparently neutral, may generate deposits or encrustations in the primary element. The consequent changes in the coefficient of discharge which may result from these deposits or encrustations can lead to values outside the tolerances given in this standard.

5.1.3 The primary element shall be manufactured from material the coefficient of thermal expansion of which is known.

5.2 Type of Fluid — The term fluid when used in this standard shall mean petroleum gases and town gases only which should be physically and thermally homogeneous at the point of measurement.

5.3 Installation

5.3.1 The measuring process applies only to fluids flowing through a pipe.

5.3.2 The primary element is fitted between two sections of straight cylindrical pipes of constant cross-sectional area, in which there is no obstruction or branch connection (whether or not there is flow into or out of such connections during measurement) other than those specified in this standard. The pipe is considered straight when it appears so by visual inspection.

5.3.3 The pipe bore is truly circular over the entire minimum lengths of straight pipe required. The cross-section is taken to be circular if it appears so by mere visual inspection. The circularity of the outside of the pipe may be taken as a guide, except in the immediate vicinity of the primary element. Over an upstream length of at least $2D$ measured from the upstream face of the primary element, the pipe shall be cylindrical. The value of the diameter D of the pipe shall be taken as the mean of the measurements of several diameters situated in meridian planes at approximately even angles to each other and in several planes normal to the pipe centre line within the specified length of $2D$. At least 4 diameters shall be measured.

5.3.4 The inside diameter of the pipe shall be equal to or more than 50 mm and equal to or less than the maximum diameters specified for each device.

5.3.5 The inside surface of the pipe immediately preceding the primary element shall be clean, free from pitting and deposit and not encrusted.

5.3.6 The rate of flow shall be constant or, in practice, vary only slightly and slowly with time. This standard does not provide for the measurement of pulsating flow.

5.3.7 The flow of fluid through the primary element shall not cause any change of phase. The computation of flow shall be carried out on the assumption that the expansion is isentropic.

5.3.8 The ratio of the downstream to the upstream absolute pressure shall be greater than 0·75.

5.3.9 The straight lengths given in Table 1 (for orifice plates and nozzle including venturi nozzle) and Table 2 (for venturi tubes) are minimum values, and it is always recommended to have straight lengths longer than those indicated. For research work especially, it is recommended to double at least the upstream values given in Tables 1 and 2, for zero additional tolerance.

5.3.10 At the time of installation of a primary element it is necessary to assure that the limitations on β, Re_D, D, $\frac{P_2}{P_1}$, k and $\frac{D}{k}$ as specified in Table 3 are strictly observed. It should also be noted that the unbracketed terms in Table 3 are the extreme values and this standard recommends that the values within brackets should be used for all practical purposes.

5.3.11 The valves mentioned in Table 1 and Table 2 should be fully open. It is recommended that control be effected by valves located downstream of the primary element. Isolating valves located upstream shall be preferably of the gate type and shall be fully open.

5.3.12 After a single change of direction (bend or tee), it is recommended that the tappings (if single tapping) be in a plane at right angles to the plane containing the change of direction.

5.3.13 The values given in Table 1 and Table 2 have been obtained experimentally with a very long straight length upstream of the particular fitting in question. Usually, such conditions are not available and the following remarks may be used as a guide in usual installation practice:

a) If the primary element is installed in a pipe leading to an upstream open space or large vessel, either directly or through any fitting given in Table 1 or 2, the total length of pipe between the open space and the primary element should never be less than 30 *D*.

b) If several fittings other than 90° bends (in the case of several 90° bends, refer to Table 1 or 2 which can be applied whatever the length between two consecutive bends) are placed in series upstream from the primary element, the following rule should be applied between the closest fitting (1) to the primary element and the primary element itself, there should be a minimum straight length such as is indicated for the fitting (1) in question and the actual value β in Tables 1 or 2. But, in addition, between this fitting (1) and the preceding one (2) there should be a straight length equal to one half of the value given in Table 1 (for orifice plates and nozzles) and Table 2 (for venturi tubes and venturi nozzles) for fitting (2) applicable to a primary element of diameter ratio $\beta = 0{\cdot}7$ (whatever the actual value of β may be). This requirement does not apply when the fitting (2) is an abrupt symmetrical reduction, which case is covered by paragraph (a) above.

5.3.14 For metering gas, the primary element shall be installed at a point where pressure is lowest and the temperature is the highest in order to minimize gas law deviations. However, when possible, the pressure should be high enough so that small changes in pressure drop across the primary element have an inappreciable effect on the expansion factor ξ and so that the pressure loss through the primary element is only a small fraction of the power requirements for pumping.

5.3.15 The minimum line size required for the installation of a primary element is given in Table 3. Where the line size is less than the specified minimum value, it should be increased at least to the minimum value for the metering run.

5.3.16 All primary elements shall be inspected periodically. For intermittent or batch units or for the units having frequent shutdowns, the primary element can be removed during idle periods but on continuous operating units special provision shall be made for removing important primary elements. By-pass connections can be made around the primary element; or special fittings, so designed as to allow removal of the plate during operation can be used. By-passes have the disadvantage of high cost, since double block valves with a bleeder valve between them should be installed in by-pass lines to ensure that no fluid flows through the by-pass when the primary element is in service. If by-passes are used, care should be taken that the section containing the primary element meets the necessary requirements of straight pipe lengths preceding and following the element.

5.4 Special Requirements for Orifice Meters and Nozzles

5.4.1 The minimum upstream and downstream straight lengths of pipe required for installation between various fittings and the primary element vary according to the nature of the fitting and are given in Table 1.

5.4.2 The mean diameter of the downstream straight length, considered along a length of at least $2D$ from the upstream face of the primary element, should not differ from the mean diameter of the upstream straight length by more than ± 2 percent, this being judged by the check of a single diameter of the downstream straight length.

5.5 Special Requirements for Venturi Tubes and Venturi Nozzles

5.5.1 The mean diameter of the pipe D where it joins the classical venturi tube shall be within ± 1 percent of the classical venturi tube entrance cylinder diameter. Moreover, no single diameter of this inlet pipe section shall differ from the mean of diameters by more than ± 2 percent for a distance of two pipe diameters preceding the classical venturi tube.

The diameter of the pipe immediately downstream of the venturi tube need not be measured accurately but it should be checked that, if the length of the divergent is less than four times the throat diameter, the downstream pipe diameter is not less than 90 percent of the diameter at the end of the venturi tube divergent. This means that, in most cases, pipes having the same nominal bore as that of the venturi tube may be used.

5.5.2 In the case of the venturi nozzle, the inner diameter d' of the upstream piezometer ring shall be between the pipe diameter D and $1{\cdot}02\ D$, and its axial length shall be smaller than $0{\cdot}2\ D$. The diameter ratio β shall be calculated using the pipe diameter D and not the diameter d' of the ring.

5.5.3 The minimum straight lengths of pipe to be installed upstream of the classical venturi tube and following the various fittings are given in Table 2. These values are expressed in multiples of D and shall be measured from the plane of the upstream pressure taps of the classical venturi tube. The pipe surface, at least for the minimum length given in Table 2, shall have a roughness not greater than that of commercially smooth pipe (approximately $k < 0{\cdot}2$ mm).

5.5.4 Fittings or obstructions, listed in Table 2, placed at least four throat diameters downstream from the throat pressure tap plane do not affect the accuracy of measurement.

5.5.5 For venturi nozzles the data given for the values of flow coefficient (*see* **12.6**) were obtained with an upstream straight pipe length of more than 50 D. The minimum straight lengths to be installed upstream of nozzles (*see* **5.4.1**) may be satisfactory for venturi nozzles.

5.6 The primary element shall be calibrated under actual installation conditions in cases which are not covered by **5.1** to **5.5**.

6. SYMBOLS AND DEFINITIONS

6.1 Symbols used in this standard are given in Table 4.

6.2 Pressure Measurement — Definitions in this standard are given only for terms used with a special meaning.

6.2.1 *Pipe-Wall Pressure Tap* — Hole drilled in the wall of a pipe, the inside edge of which is flush with the inside surface of the pipe. The hole is usually circular but in certain cases may be an annular slit.

6.2.2 *Static Pressure of a Fluid Through a Straight Pipeline* — Pressure which can be measured by connecting a pressure gauge to a pipe-wall pressure tap. Only the value of the absolute static pressure is used in this standard.

6.2.3 *Differential Pressure* — Difference between the static pressure measured by pipe-wall taps, one of which is on the upstream side and the other on the downstream side of a primary element inserted in a straight pipe through which flow occurs, when there is no variation in gravitational energy between the upstream and downstream taps. The term differential pressure is used only if the pressure taps are in the positions specified in the specification for each standard primary element.

6.2.4 *Differential Pressure Ratio* — The differential pressure divided by the absolute static pressure existing at the level of the intersection of the centre-line of the primary element with the upstream pressure tap plane.

6.2.5 *Pressure Loss*

a) *For orifice plates and nozzles* — It is the difference in static pressure between the pressure measured on the upstream side of the primary element, at a point free from the influence of approach impact pressure, and that measured on the downstream side of the element, at a point where static pressure recovery by expansion of the jet is completed.

b) Pressure loss caused by venturi tube (*see* Fig. 1) may be determined by pressure measurements made prior and subsequent to the installation of the venturi tube in a pipe through which there is a given flow. If p'' is the difference in pressure, measured prior to the installation of the venturi tube, between two pressure taps one of which is situated at least $1D$ upstream of the flanges where the venturi tube should be inserted and the other is $6D$ downstream of the same flanges, and if p' is the difference in pressure measured between the same pressure taps after installation of the venturi tube between these flanges then the pressure loss caused by the venturi tube is given by $p' - p''$.

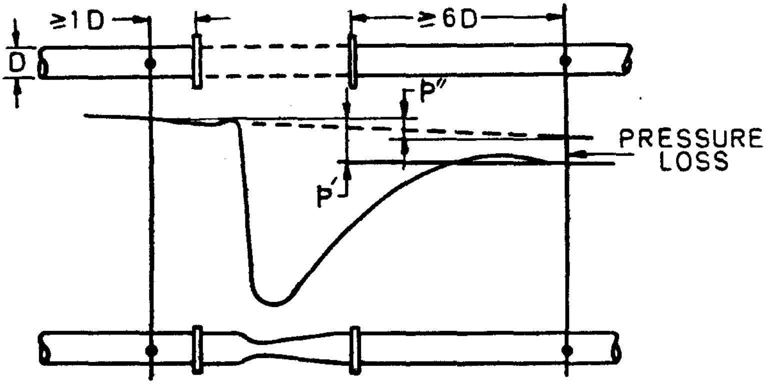


FIG. 1 PRESSURE LOSS ACROSS VENTURI TUBE

6.3 Primary Elements

6.3.1 *Orifice or Throat* — Opening of minimum cross-sectional area in a primary element. Standard primary element orifices are always circular and coaxial with the pipe line.

6.3.2 *Orifice Plate* — Thin plate in which a circular aperture has been machined. Standard orifice plates are described as thin plate and with sharp square edge, because the thickness of the plate is small compared with the diameter of the measuring section and the upstream edge of the orifice is sharp and square.

6.3.3 *Nozzle* — Device which consists of a convergent inlet to a cylindrical portion generally called the 'throat'.

6.3.4 *Venturi Tube* — A venturi tube has a convergent part connected to a cylindrical part called the 'throat' and an expanding section called the 'divergent' which is conical for normal types. If the convergent part is conical, the device is called a 'classical venturi tube'. If the convergent part is a standard nozzle, the device is called a 'venturi nozzle'.

There are three methods of constructing classical venturi tubes which are characterized particularly by the method of manufacture of the inside surface of the convergent part and the amount of rounding at the intersection of the convergent part and the throat. These three construction methods give them slightly different flow characteristics.

The three varieties of classical venturi tubes are as follows:

a) *Classical venturi tube with a rough-cast convergent part* — Classical venturi tube made by casting in a sand mould or by other methods which leave a finish on the surface of the convergent part similar to that of the sand casting. The throat should be machined and the junctions between the cylinders and cones should be rounded.

b) *Classical venturi tube with a machined convergent part* — Classical venturi tube cast or fabricated as above but in which the convergent part is machined, as are the throat and entrance cylinder. The junctions between the cylinders and cones may or may not be rounded.

c) *Classical venturi tube with a rough welded sheet-iron convergent part* — Classical venturi tube which is normally fabricated by welding and which is not machined in any way in the larger sizes, but which has the throat machined in the smaller sizes.

6.3.5 Diameter ratios of a primary element in a given pipe are defined as follows:

a) For orifice plates and nozzles it is the diameter of the orifice of the primary element divided by the diameter of the measuring pipe upstream of the primary element.

b) When the primary element has a cylindrical section upstream, equal in diameter to that of the pipe as in the case of the classical venturi tube, the diameter ratio is the quotient of the throat diameter at the plane of the pressure taps by the diameter of this cylindrical section at the plane of the upstream pressure taps (*see* also **11.2.1** and **11.2.3**).

c) The diameter ratio of a venturi nozzle used in a given pipe is the quotient of the diameter of the throat of the venturi nozzle by the mean diameter of the pipe upstream of this venturi nozzle.

6.4 Flow

6.4.1 *Rate of Flow of Fluid Passing Through a Primary Element* — is the quantity of fluid passing through a primary element in unit time. This quantity can be characterized by its mass or its volume at specified temperature and pressure and the rate of flow can be expressed in units of mass or volume per unit time. In all cases, it is necessary to state explicitly whether the type of flow rate referred to is expressed by mass or by volume per unit time.

6.4.2 *Pipe Reynolds Number* — The pipe Reynolds number used in this standard is referred to the upstream condition of the fluid and to the upstream diameter of the pipe, that is $Re_D = \frac{V_1 D}{\nu_1}$.

6.4.3 *Isentropic Exponent* — The isentropic exponent κ appears in the different formulae for expansibility (expansion) factor ξ either directly or in the ratio X. There are many gases and vapours for which no values for κ have been published so far. For gases, however, the behaviour of which fairly equals that of ideal gases, the isentropic exponent may be replaced by the ratio of the specific heat capacities. The isentropic exponent, as well as the ratio of the specific heat capacities, vary in general whenever the gas temperature and/or pressure vary.

6.4.4 *Acoustic Ratio* — The differential pressure ratio divided by the isentropic exponent (compressible fluid).

6.4.5 *Velocity of Approach Factor* — It is equal to:

$$E = (1 - \beta^4)^{-\frac{1}{2}} = D^2/\sqrt{D^4 - d^4} = (1 - m^2)^{-\frac{1}{2}}$$

6.4.6 *Flow Coefficient* — Calibration of standard primary elements by means of incompressible fluids (liquids) shows that the quantity α defined by the following relation is dependent only on the Reynolds number for a given primary element in a given installation:

$$\alpha = \frac{q_m}{\frac{\pi}{4} d^2 \sqrt{2 \Delta P \rho_1}}$$

The quantity α, a pure number, is called the 'flow coefficient'. The numerical value of α is the same for different installations, whenever such installations are geometrically similar and the flows are characterized by identical Reynolds number.

The ratio $C = \frac{\alpha}{E}$ is called the 'coefficient of discharge'.

The numerical values of α and of C given in this standard have been determined experimentally.

6.4.7 *Expansibility (Expansion) Factor* — Calibration of a given primary element by means of a compressible fluid (gas) shows that the ratio

$$\frac{q_m}{\frac{\pi}{4} d^2 \sqrt{2 \Delta P \rho_1}}$$

is dependent both on the value of the Reynolds number and on those of the relative differential pressure and the isentropic exponent of the gas. The method adopted for representing these variations consists in multiplying the flow coefficient α of the considered primary element, as determined by direct calibration effected by means of liquids for the same value of Reynolds number, by the 'expansibility', a so-called (expansion) factor defined by the relation:

$$\epsilon = \frac{q_m}{\frac{\pi}{4} \alpha d^2 \sqrt{2 \Delta P \rho_1}}$$

ϵ differs from and is less than unity, when the fluid is compressible.

This method is possible because experiments show that practically ϵ is independent of Reynolds number, and, for a given diameter ratio of a given primary element, depends on the differential pressure ratio and the isentropic exponent.

The numerical values of ϵ given in this standard have been determined experimentally.

6.4.8 *Pressure Ratio* — The ratio of the absolute pressure at the throat taps to that at the upstream taps. It is thus equal to $(1 - x)$.

7. TEMPERATURE, PRESSURE, VISCOSITY AND DENSITY EVALUATION

7.1 Temperature Measurement — In setting up a meter installation, a means of determining the flow temperature shall be provided. Measurements are usually made with thermocouples or mercury in glass thermometers, which are placed in thermowells correctly located with respect to the primary element. The presence of thermowells creates flow disturbance which are suppressed, before they reach the primary element, by installing wells a sufficient distance upstream or downstream. In either case, the following specifications are recommended:

a) On the upstream side, thermowells shall precede the primary element by at least 20 D. If straightening vanes are provided, thermowells shall be placed at least 10 D upstream from the inlet edge of the vanes,

b) On the downstream side, thermowells shall not be located closer than 5 D to the primary element.

For gas flow, pressure and temperature conditions shall be evaluated on the same side of the primary element. However, if the pressure drop across the primary element is small compared to the total pressure, the change in temperature due to expansion will be negligible; and the temperature measured on either side will be sufficiently accurate. On the other hand, when the pressure drop is greater than 0·4 P_1 and $\frac{P_2}{P_1} \leqslant 0{\cdot}60$, the thermowell shall be located on the upstream side of the primary element.

7.2 Pressure Measurement — In calculating flow rates for compressible fluids, it is necessary to know the static pressure of the flowing stream. Measurements are usually recorded on a meter-chart by a pen actuated by a pressure responsive device which is connected to the lead line from either the upstream or downstream pressure tap. Since flow equations may be set up in terms of either pressure condition, location of the static pressure pen connection appears to be a question of local preference, with a slight advantage in favour of the downstream position. If the upstream pressure is constant (for example controlled by a pressure regulator), static pressure can be obtained with an indicating gauge connected to the high pressure lead. As a general rule, however, a record of static pressure is needed in conjunction with the differential pressure record.

Even though a meter is equipped with a pressure pen, a connection for a test gauge installation shall be provided in the lead line to which the pen is connected; and the pressure reading shall be checked frequently with a calibrated test gauge. For low range pressure springs, it may be preferable to use a mercury manometer for calibration.

When pressure pens are installed, they may be adjusted to give a chart record proportional to either absolute or gauge static pressure. For example, in the case of 0-100 kg springs, the installation can be such that the zero to full scale chart reading corresponds to a range of either 0-100 kg absolute or 0-100 kg gauge. Since absolute pressure is needed for flow calculations, it is recommended that pens be set to read absolute pressure directly.

7.3 Evaluation of Viscosity — To evaluate the viscosity of the flowing gas, follow the procedure given below:

a) From a knowledge of molecular mass (*MM*) and upstream pressure (P_1) of the gas, determine reduced pressure P_R from Fig. 2.

b) Utilising upstream temperature (T_1) and molecular mass (*MM*), determine reduced temperature T_R from Fig. 3.

c) Determine dynamic viscosity η_1 of the gas at operating temperature and 1 atmospheric pressure from Fig. 4.

d) Evaluate from Fig. 5 the pressure correction factor corresponding to P_R and T_R.

e) Multiply η_1 with this pressure correction factor to get the actual dynamic viscosity η for the gas at operating conditions.

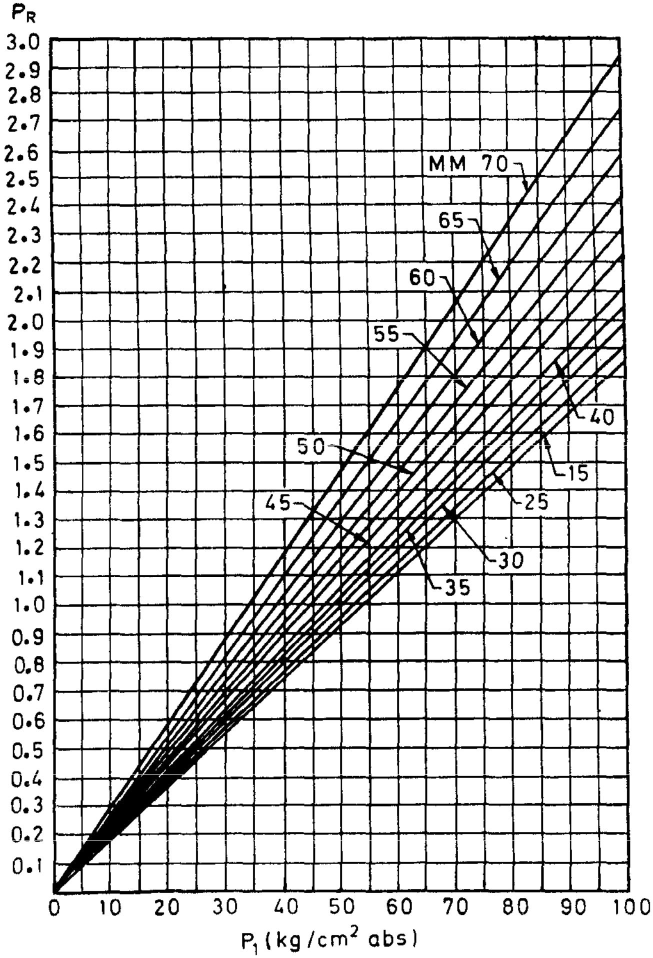


FIG. 2 ESTIMATED REDUCED PRESSURE OF HYDROCARBON GASES

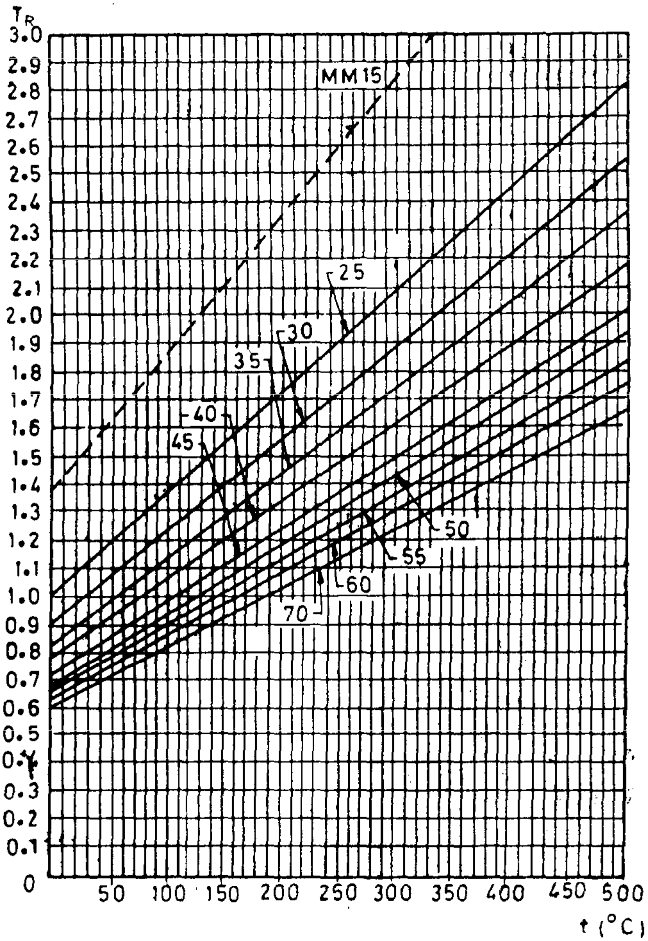


Fig. 3 Estimated Reduced Temperature of Hydrocarbon Gases

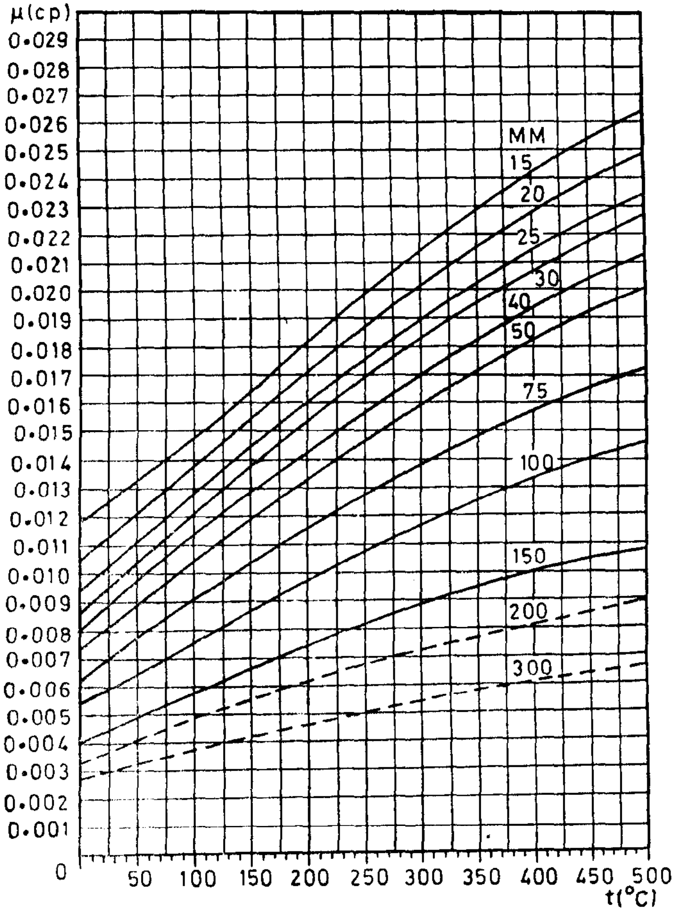
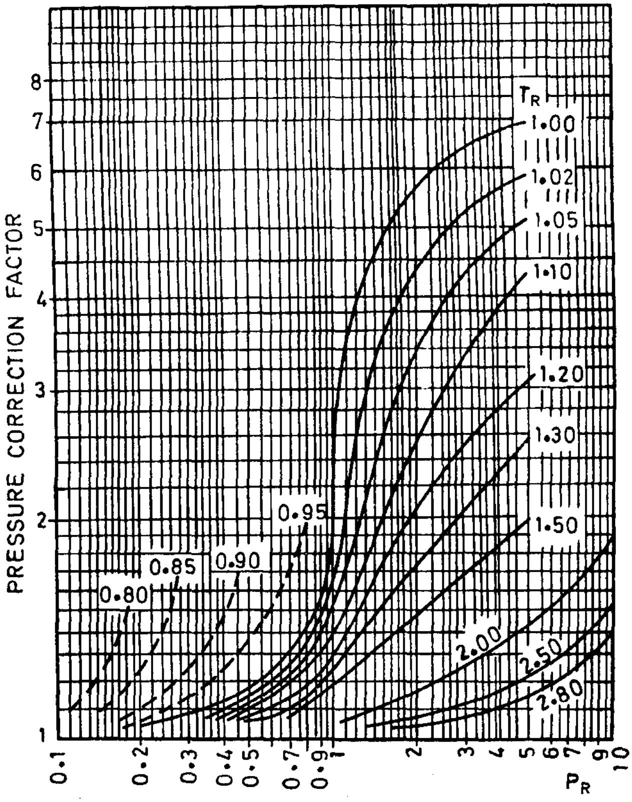


Fig. 4 Dynamic Viscosity of Mixtures of Hydrocarbon Gases at Atmospheric Pressure



NOTE—For P_R and T_R *see* Fig. 2 and 3.

To determine the dynamic viscosity of gases at higher pressures, multiply the viscosity at atmospheric pressure and actual temperature by the pressure correction factor obtained from this graph.

FIG. 5 DYNAMIC VISCOSITY OF GASES, PRESSURE CORRECTION FACTOR

7.4 Evaluation of Density — Density of the gas at flow temperature and pressure conditions shall be evaluated from the formula

$$\rho_1 = \frac{M P_1}{R T Z}$$

where

M = molecular mass of the gas,

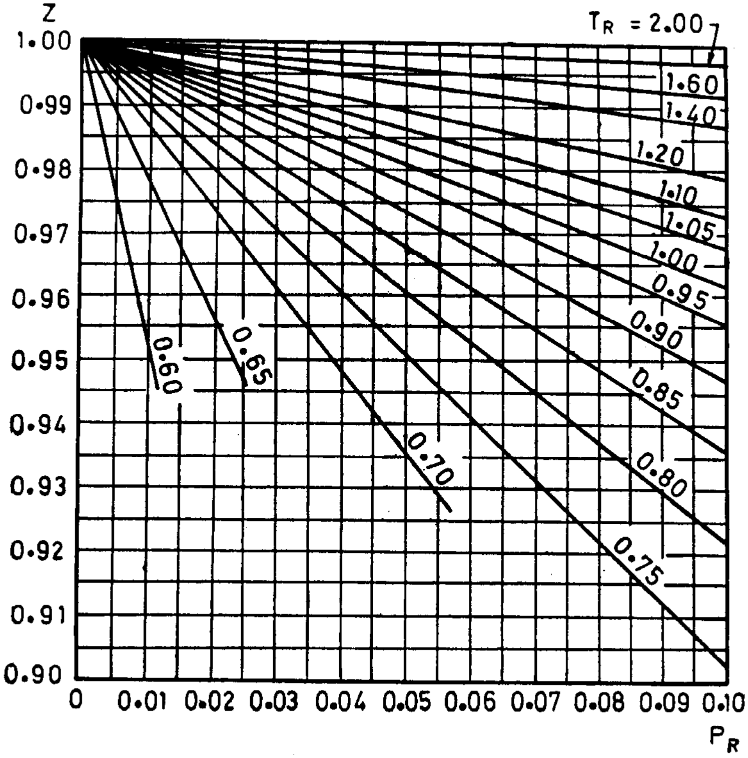
P_1 = pressure of the gas in kgf/cm²,

R = universal gas constant which is equal to 84·8 $\frac{\text{kg cm}}{\text{mol °K}}$,

T = absolute temperature of the gas in °K,

Z = compressibility factor which is equal to $\frac{P_1 \overline{V}}{R T}$, and

$\overline{V}$ = volume in cm³ of 1 mole of gas under the prescribed conditions.



NOTE—Graph valid for most gases and gas mixtures with an error < 1%. Do not use for H_2, He, H_2O, NH_3, and gas mixtures with H_2.

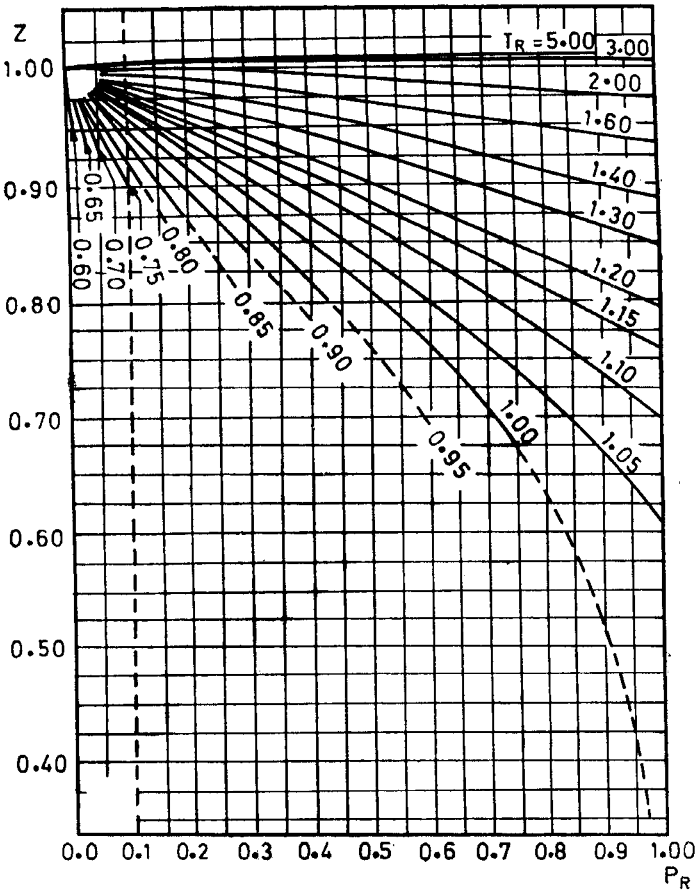
FIG. 6A COMPRESSIBILITY FACTORS FOR GASES (P_R UP TO 0·10)

The compressibility factor usually increases as:

a) the pressure is increased,
b) the temperature is decreased, and
c) the molecular mass is increased.

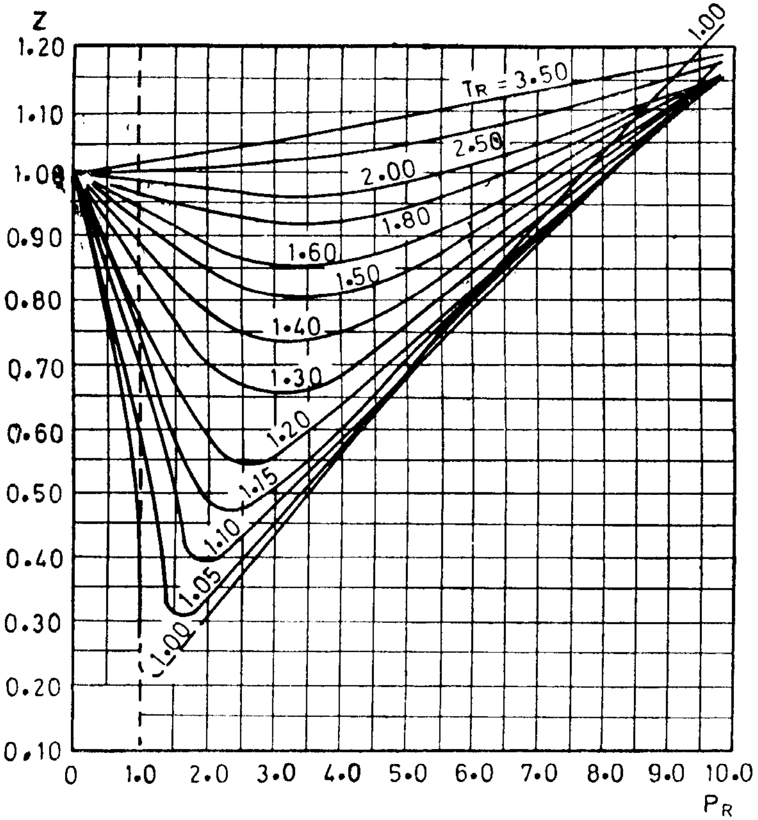
The procedure to determine compressibility factor for hydrocarbon gases is as follows:

a) Determine the estimated reduced pressure (P_R) and reduced temperature (T_R) of the hydrocarbon gas from Fig. 2 and 3 respectively; and
b) Evaluate the compressibility factor from Fig. 6.



NOTE — Graph valid for most gases and gas mixtures with an error < 1%. Do not use for H_2, He, H_2O, NH_3 and gas mixtures with H_2.

FIG. 6B COMPRESSIBILITY FACTORS FOR GASES (P_R UP TO 1·0)



NOTE — Graph valid for most gases and gas mixtures with an error < 2·5%. Do not use for H_2, He, H_2O, NH_3, methyl fluoride and gas mixtures with H_2.

FIG. 6C COMPRESSIBILITY FACTORS FOR GASES (P_R UP TO 10·0)

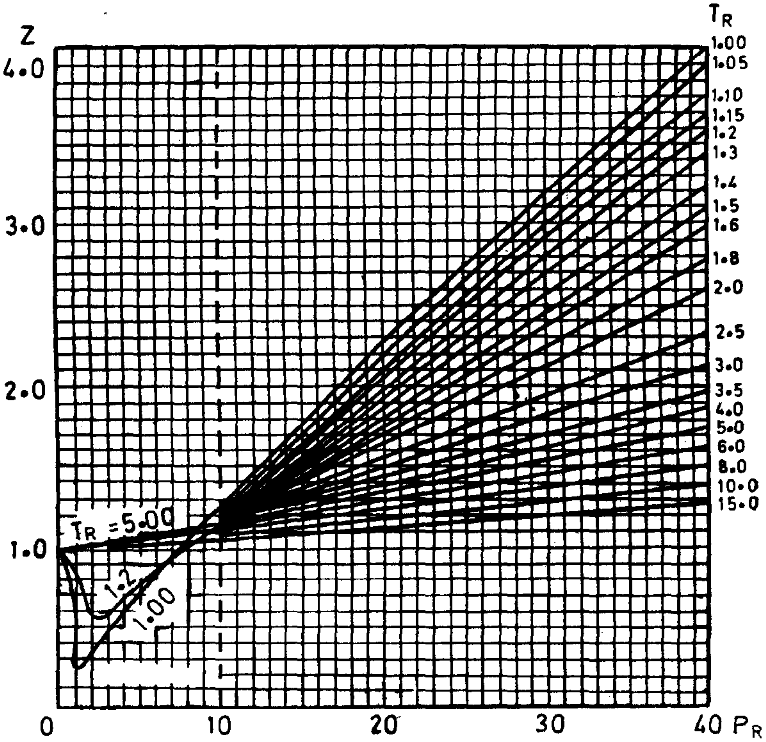
8. COMPUTATION FORMULAE

8.1 Basic Formula

8.1.1 For calculating the mass rate of flow, q_m, the flow coefficient α and expansibility (expansion) factor ϵ as specified in this standard, shall be used in the following formula:

$$q_m = \alpha \, \epsilon \, \frac{\pi}{4} \cdot d^2 \sqrt{2 \Delta P \, \rho_1}$$

NOTE—In this equation ΔP is expressed in $\frac{\text{kg.m}}{\text{sec}^2 \, \text{m}^2}$, d in metres, and ρ_1 in $\frac{\text{kg}}{\text{m}^3}$ to obtain q_m in kg/s.



NOTE —Graph valid for most gases and gas mixtures with an error < 5%. Do not use for H_2, He, H_2O, NH_3, methyl fluoride and gas mixtures with H_2.

FIG. 6D COMPRESSIBILITY FACTORS FOR GASES (P_R UP TO 40)

8.1.2 Similarly, the value of the volume rate of flow, at the upstream conditions of the fluid, may be calculated by the following relation:

$$q_{v1} = q_m/\rho_1$$

8.1.3 The formulae of **8.1.1** and **8.1.2** apply for any consistent system of units.

8.1.4 The diameter of throat or orifice of the primary element d at operating conditions can be determined from a knowledge of the throat or orifice diameter 'd_m' at any specified temperature t_m, and the coefficient of thermal expansion, θ_m, for the material from which the primary element has been manufactured. This is computed from the equation

$$d = d_m \left[1 + \theta_m \left(t - t_m\right)\right]$$

where

d and d_m are expressed in mm,

t and t_m are expressed in °C, and θ_m is expressed in $\frac{\text{mm}}{\text{mm°C}}$

Table 5 gives values of θ_{m} for various substances.

8.2 Method of Determination of a Standard Primary Element — The principle of the method consists essentially in selecting *a priori*:

a) the type of standard primary element to be used, and

b) a rate of flow and the corresponding value of the differential pressure.

The related values of q_{m} and $\triangle P$ should be inserted in the basic formula rewritten in the form below:

$$\alpha\beta^2 = \frac{4\, q_{\mathrm{m}}}{\epsilon\, \pi\, D^2 \sqrt{2 \triangle P \rho_1}}$$

and the diameter ratio of the selected primary element is determined by successive approximations.

8.3 Computation of the rate of flow is effected by replacing the different terms on the right hand side of the basic formula:

$$q_{\mathrm{m}} = \alpha\, \epsilon\, \frac{\pi}{4} d^2 \sqrt{2 \triangle P \rho_1}$$

by their numerical values, obtained in the course of the measurement, and by calculating their product. The computation itself involves no difficulty other than of an arithmetical nature and merely calls for the following comments:

a) α may be dependent on Re_{D}, which is itself dependent on q_{m}. Therefore, the final value of q_{m} may be obtained by successive approximations, after first calculating q_{m} from a value of Re_{D} (or of α) chosen *a priori*. For instance, $\alpha = \alpha_0$ can be taken as a first value.

b) $\triangle P$ represents the differential pressure as defined under **6.2.3.** If the difference in pressure as measured by the secondary device does not conform to this definition, the value of the differential pressure is calculated on the basis of the reading given by the secondary device, taking account of the circumstances of its use, in order, for instance, to account for possible changes in gravitational energy between upstream and throat taps.

9. ORIFICE PLATES

9.0 The various types of standard orifice plates are similar and therefore need only one description. Each type of standard orifice plate is characterized by the arrangement of pressure taps. The limitations for the use of square edge orifice with different types of pressure taps are given in Table 3. No orifice plate can be used according to this standard unless it conforms with the following requirements.

9.1 Description — The meridian plane cross section of the plate is shown in Fig. 7. The letters in the figure are for reference purposes.

9.1.1 *General Shape* — The part of the plate inside the pipe shall be circular and concentric with the pipe centre-line. The faces of this plate shall always be flat and parallel. Unless otherwise stated, the following descriptions apply only to that part of the plate intended to be located within the pipe for measuring purposes.

9.1.2 *Upstream Face* A

9.1.2.1 The upstream face of the plate A shall be flat. It is considered as such when the slope of a straight line connecting any two points of its surface in relation to a plane perpendicular to the centre-line is less than 1 percent, ignoring the inevitable local defects of the surface which are invisible to the naked eye.

9.1.2.2 The upstream face of the orifice plate shall be smooth to within $0{\cdot}0003\ d$ (peak-to-hollow height) within a circle whose diameter is not less than $1{\cdot}5\ d$ and which is concentric with the orifice.

9.1.2.3 It is useful and convenient to provide a distinctive mark which, whenever possible, is visible even when the orifice plate is installed to show that the upstream face of the orifice plate is correctly installed relative to the direction of flow.

9.1.3 *Downstream Face* B

9.1.3.1 The downstream face B shall be flat and parallel with the upstream face.

9.1.3.2 The flatness and surface condition of the downstream face are judged by mere visual inspection (*see* **9.1.2.1** and **9.1.4.4**).

9.1.4 *Thickness* E *and* e

9.1.4.1 The thickness e of the orifice shall be between $0{\cdot}005\ D$ and $0{\cdot}02\ D$; however, when the value of β is less than $0{\cdot}2$, the thickness e shall be between $0{\cdot}005\ D$ and $0{\cdot}1\ d$ (this latter condition does not apply when corner taps are used, because the lower limit for β is then $0{\cdot}22$).

9.1.4.2 The values of e measured at any point on the orifice shall not differ among themselves by more than $0{\cdot}001\ D$.

9.1.4.3 The thickness E of the plate shall be between e and $0{\cdot}05\ D$.

9.1.4.4 The values of E measured at any point of the plate shall not differ among themselves by more than $0{\cdot}005\ D$.

9.1.4.5 For orifice plates with corner taps and when $\beta > 0{\cdot}7$, e shall be approximately equal to one-third of E, if the plate is bevelled.

9.1.5 *Angle of Bevel* F

9.1.5.1 If the thickness E of the plate exceeds the thickness e of the orifice, the plate shall be bevelled on the downstream side. The bevelled surface shall be well finished.

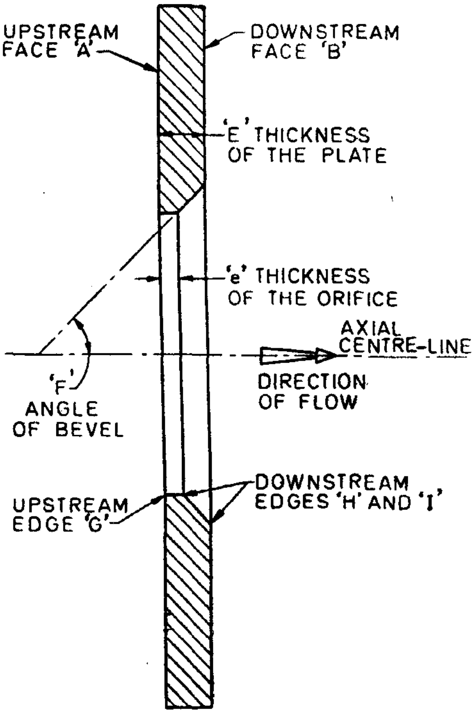


FIG. 7 STANDARD ORIFICE PLATE

9.1.5.2 The angle of bevel F shall be between 30° and 45°.

9.1.5.3 The plate may be not bevelled if its thickness E is less than or equal to 0·02 D (*see* **9.1.4.1**).

9.1.6 *Edges* G, H *and* I

9.1.6.1 The upstream edge G and the downstream edges H and I shall have neither wire-edges nor burrs, nor, in general, any peculiarities visible to the naked eye.

9.1.6.2 The upstream edge G shall be sharp. It is considered so if, when viewed with the naked eye, it does not seem to reflect a beam of light.

9.1.7 *Diameter of Orifice* d

9.1.7.1 The value d of the diameter of the orifice shall be taken as the mean of the measurements of a number of diameters distributed in meridian planes and at approximately even angles to each other. At least four diameters shall be measured.

9.1.7.2 The orifice shall be cylindrical. No diameter shall differ by more than 0·05 percent from the value of the mean diameter. This requirement is satisfied when the difference in the length of any of the measured diameters complies with the said requirement in respect of the mean of the measured diameters.

9.1.7.3 The diameter d shall always be between 0·1 D and 0·8 D, although this range does not cover certain standard orifice plates. The choice of the ratio d/D, within these limits, is left to the user; it is one of the parameters which defines an orifice plate of a given type.

9.1.8 *Symmetrical Plates*

9.1.8.1 If the orifice plate is intended to be used for measuring reverse flows :

a) the plate shall not be bevelled;

b) the two faces shall both be as described for the upstream face; the thickness E of the plate shall be equal to the thickness e of the orifice; and

c) the two edges of the orifice shall be as described for the upstream edge.

9.1.8.2 Furthermore, for orifice plates with vena contracta taps, two sets of upstream and downstream pressure taps may be arranged and used alternately (*see* **9.2**).

9.1.9 *Material and Manufacture*

9.1.9.1 The plate may be manufactured of any material and in any way, provided it is and remains in accordance with the foregoing description during the proposed flow measurements. In particular, the plate shall be clean when the measurements are made.

9.1.9.2 The orifice plate is usually made of metal, and preferably should be erosion-proof and corrosion-proof.

9.1.9.3 The machining required to obtain a plate conforming to the standard description usually calls for the use of machine tools of good quality and in good condition.

9.1.9.4 If the plate is turned, the best result is obtained if, after the orifice has been bored, the upstream face is finished by a very fine radial cut from the centre outwards.

9.2 Pressure Taps — For all types of standard orifice plates, pressure taps shall be as defined in **6.2.1**. At least one upstream pressure tap and one downstream pressure tap shall be provided for each primary element, installed in one or other of the recommended standard positions. A single plate can be used with several sets of pressure taps suitable for different types of standard orifice plates.

9.2.1 *Relative Advantages and Limitations of Various Tap Connections* — It is desirable for several reasons to locate the pressure taps at a point where the change in pressure is small:

a) because there is less turbulence, and
b) because the exact location and size of the tap hole are less important at the point of steady pressure.

The relative advantages and limitations of various tap connections are given in **9.2.1.1**, **9.2.1.2** and **9.2.1.3**.

9.2.1.1 *Corner taps* — These taps are more susceptible to dirt, freezing and clogging with hydrates when measuring natural gas and more dependent on the thickness of plate and gasket than other tap connections. Corner taps are more affected by upstream disturbances and generally less reliable than flange or vena contracta taps.

9.2.1.2 *Vena contracta taps* — From **9.2.1** it can be concluded that vena contracta taps (downstream tap at point of minimum pressure) as well as the type with taps 2·5 *D* upstream and 8 *D* downstream (downstream tap at point of full pressure recovery these are also called pipe taps) are best. But this slight theoretical advantage is counterbalanced by many practical considerations. Certainly tap locations at points of extreme pressure instability (such as between 0·75 and 3 *D* downstream) should be avoided, but the difference between the stability at the vena contracta and at the downstream flange tap location is small.

9.2.1.3 *Flange taps* — These are probably the most commonly used tap connections. These have the following advantages:

a) They are located close to the face of the flange where they are accessible for inspection;
b) They are located by the manufacturer of the flange and thus are largely independent of the human element; and
c) They are symmetrical and hence adapt themselves to measurement of flow in either direction.

9.2.2 *Shape and Diameters of Pressure Taps*

9.2.2.1 Corner taps are dealt with in **9.2.5**.

9.2.2.2 The centre-line of the tapping shall meet the pipe centre-line and be at right angles to it.

9.2.2.3 At the point of breakthrough the hole shall be circular. The edges shall be flush with the pipe-wall, free from burrs and generally have no peculiarities and be rounded to a radius not exceeding one-tenth of the diameter of the pressure tap.

9.2.2.4 Conformity of the pressure taps with the two foregoing descriptions is judged by visual inspection.

9.2.2.5 The diameters of pressure taps shall be less than 0·08 *D* and preferably between 6 and 12 mm. The upstream and downstream pressure taps shall have the same diameter.

9.2.2.6 The pressure pipes to the secondary devices shall be cylindrical for a length, from the inside wall of the measured pipe, at least equal to twice the diameter of the pressure taps.

9.2.3 *Angular Position of Pressure Taps*

9.2.3.1 Centre-lines of the pressure taps may be located in any meridian sector of the pipeline. However, care shall be taken to ensure that the location prevents entry of entrained liquid condensate, etc, into the taps.

9.2.3.2 The axis of the upstream tap and that of the downstream tap may be located in different meridian sectors. However, attention is drawn to the fact that, in any case, the readings of differential pressure obtained by these pressure taps shall be in accordance with the definitions in **6.2.3**, particularly with respect to gravitational energy variations.

9.2.4 *Spacing of Pressure Taps*

9.2.4.1 The spacing of a pressure tap is the distance, measured on a straight line parallel to the centre-line of the pipeline, between the centre-line of the pressure tap and the plane of one specific face of the orifice plate.

9.2.4.2 Pressure taps spacing characterizes the type of standard orifice plate.

9.2.4.3 *Orifice plate with pipe taps*

Upstream tap — spaced 2·5 D from the upstream face of the plate.

Downstream tap — spaced 8 D from the downstream face of the plate.

9.2.4.4 *Orifice plate with vena contracta taps*

Upstream tap — spaced between 0·9 D and 1·1 D from the upstream face of the plate.

Downstream tap — spaced from the upstream face of the plate as defined in Table 6.

9.2.4.5 *Orifice plate with flange taps*

Upstream tap — spaced 25 ± 1 mm from the upstream face of the plate.

Downstream tap — spaced 25 ± 1 mm from the downstream face of the plate.

9.2.5 *Corner Taps*

9.2.5.1 The spacing between the taps and the respective faces of the plate is equal to half the diameter or half the width of the taps themselves, so that the tap holes breakthrough the pipe walls flush with the faces of the plate (*see also* **9.2.5.5**).

9.2.5.2 The pressure taps shall always be either single tappings, or annular slits, opening into the annular chambers of piezometer rings, as shown in Fig. 8 to which the following letters refer.

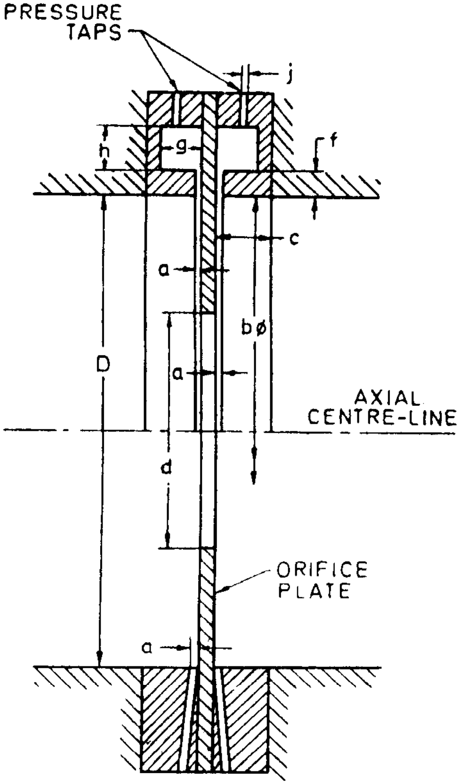


FIG. 8 PIEZOMETER RINGS FOR ORIFICE PLATES, AND CORNER TAPS (INDIVIDUAL TAPS) FOR ORIFICE PLATES

9.2.5.3 The diameter '*a*' of single taps or the width '*a*' of annular slits are given below :

For clean fluids $a \leqslant 0{\cdot}03\, D$ for $\beta \leqslant 0{\cdot}065$ and

$0{\cdot}01\, D \leqslant a \leqslant 0{\cdot}02\, D$ for $\beta > 0{\cdot}65$

Moreover for any values of β for clean fluids $1\ \text{mm} \leqslant a \leqslant 10\ \text{mm}$.

9.2.5.4 The annular slits usually breakthrough the pipe over the entire perimeter, with no break in continuity. If not, each annular chamber connects with the inside of the pipe by at least 4 openings, the axes of which are at equal angles to one another and the individual opening area of which is at least 12 mm².

9.2.5.5 If individual pressure taps, as shown in Fig. 8, are used, the centre-line of the taps should cross the centre-line of the pipe at as near a right angle (90°) as possible. If there are several individual pressure taps for one and the same upstream or downstream position, their centre-lines shall form equal angles with each other. The diameters of individual pressure taps are given in **9.2.5.3**.

The pressure lines establishing connection with the secondary devices are cylindrical from the inside wall of the measured pipe, for a length at least equal to twice the diameter of the pressure taps.

9.2.5.6 The inner diameter b of the rings shall not be smaller than D. The diameter of the rings shall be between D and $1{\cdot}02\ D$; the length c of the upstream ring shall be less than $0{\cdot}2\ D$ and the length c of the downstream ring less than $0{\cdot}5\ D$. The diameter ratio shall be calculated in relation to the pipe diameter D and not to the diameter of the ring.

The thickness f shall be more than or equal to twice the width 'a' of the annular slit. The area of the cross section of the annular chamber ($g \times h$) shall be more than or equal to half of the total area of the tapping holes connecting the chamber to the pipe.

9.2.5.7 All surfaces of the ring which will be in contact with the measured fluid shall be clean and have a good machine finish.

9.2.5.8 The pressure tappings j connecting the annular chambers to the secondary devices are pipe-wall tappings, circular at the point of breakthrough and with diameters between 4 and 10 mm.

9.2.5.9 The upstream and downstream piezometer rings are not necessarily symmetrical in relation to each other, but they shall both conform with the requirements given above.

9.3 Installation of Orifice Plate

9.3.1 *Position of Line*

9.3.1.1 Although an orifice plate may be located in either a horizontal or a vertical line, installation in a horizontal line is preferable in most cases. However, when gases contain condensable constituents, it may be desirable to locate the primary element in a vertical line with downward flow so that it will allow any condensed liquid to be blown through the orifice. In horizontal lines, such liquids may be trapped upstream from the orifice unless drainholes or blow down connections are provided.

9.3.1.2 The measuring section shall comply with the conditions stated in **5.3.2** to **5.3.5**, **5.3.9** to **5.3.14** and **5.4**.

9.3.2 *Location of Plate and Rings*

9.3.2.1 The plate shall be placed in the pipe in such a way that the fluid flows from the upstream face towards the downstream face (*see* **9.1.2.3**).

9.3.2.2 The plate shall be perpendicular to the centre-line of the pipe within ±1°.

9.3.2.3 The plate shall be centred in the pipe or, if applicable, in the piezometer rings. In all cases, the shortest distance between the centre-line of the orifice and the centre-line of the pipe on the upstream and downstream sides shall be less than $0{\cdot}015\ D\left(\frac{1}{\beta}-1\right)$.

9.3.2.4 When used, the piezometer rings (whose diameter is between D and $1{\cdot}02\ D$) shall be so centred that at no point they protrude into the pipe.

9.3.3 *Fixing and Gaskets*

9.3.3.1 The method of fixing and tightening shall be such that once the plate has been finally fixed in the proper position, it remains so. It is necessary, when holding the orifice plate between flanges, to allow for its free thermal expansion and to avoid buckling and distortion.

9.3.3.2 Gaskets, if used, shall be made and inserted in such a way that they do not protrude at any point inside the pipe. They shall be as thin as possible. Gasket thickness shall not exceed 3 mm, and shall be of the same thickness on both sides of the orifice plate.

9.3.3.3 The gaskets if used between the plate and the annular chamber rings shall not protrude inside the said chambers.

9.3.3.4 Gaskets shall be changed at each change or checking of the orifice plate.

9.3.4 *Static Pressure Measurement and Temperatures Measurement*

9.3.4.1 The static pressure of the fluid shall be measured in the plane of the upstream pressure tap, by means of a pipe-wall pressure tap, as described in **9.2.2**. This tap may be separate from the tap provided for measuring the upstream component of the differential pressure, unless the intention is to measure upstream and downstream pressure separately.

If there is an upstream annular chamber, the static pressure shall be measured at the wall of this annular chamber. The static pressure value to be used in subsequent computations is that existing at the level of the centre of the upstream measured cross section, which may differ from the pressure measured at the wall.

9.3.4.2 The temperature of the fluid shall preferably be measured downstream of the primary element, and the thermometer pocket shall take up as little space as possible. The distance between it and the orifice plate shall be at least equal to 5 D, if the pocket is located downstream, and in accordance with Table 1 if the pocket is located upstream.

9.3.4.3 Any method of determining the static pressure and the temperature of the fluid is acceptable if it enables a reliable value of the temperature, the viscosity and the mass density of the fluid upstream of the orifice plate to be obtained without disturbing the flow measurement in any way. The temperature of the orifice plate and that of the fluid upstream of the orifice plate are assumed to be the same.

9.3.5 *Thermal Insulation* — The measuring section and the pipe flanges shall be lagged over at least the whole length of the required straight runs. It is, however, unnecessary to lag the pipe when the temperature of the fluid, between the inlet of the minimum straight length of the upstream pipe and the outlet of the straight length of the downstream pipe, does not exceed any limiting value selected by the user, as being sufficient for the accuracy of flow measurement which he requires.

9.4 Coefficients of Orifice Plates with Corner Taps

9.4.1 *Coefficients* — The coefficients are set out in the form of tables and the method of use is given in **9.4.1.1** to **9.4.1.3**. Interpolation by proportion can be made, when necessary. Extrapolation shall never be made.

9.4.1.1 Flow coefficient α is given as follows:

$$\alpha = \alpha_0 \, r_{Re}$$

The values of α_0 are given as a function of β^4 and Re_D in Table 7.

9.4.1.2 *Roughness correction factor* r_{Re} — The correction factor r_{Re} for pipe roughness depends on the relative roughness k/D and on the pipe Reynolds number Re_D. It is computed by the following formula:

$$r_{Re} = (r_0 - 1)\left[\frac{\log_{10} Re_D}{6}\right]^2 + 1$$

with the exception that, if $Re_D \geqslant 10^6$, then one will set $r_{Re} = r_0$. The values of r_0 are given as a function of β^2 and D/k in Table 8.

9.4.1.3 Expansibility (expansion) factor ϵ is given in Table 9 as derived from the empirical formula:

$$\epsilon = 1 - (0{\cdot}3707 + 0{\cdot}3184\beta^4)\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{1}{\kappa}}\right]^{0{\cdot}935}$$

This formula is applicable only for the values of β, D and Re_D, as stated in the note under Table 7. Test results for determination of ϵ are known for natural gas, steam and air only. However, there is no objection to using the same formula for other petroleum or town gases, the isentropic component of which is known.

Moreover, it is applicable only if $\frac{P_2}{P_1} \geqslant 0{\cdot}75$.

9.5 Coefficients of Orifice Plates with Vena-Contracta Taps or Pipe Taps

9.5.1 *Coefficients* — The coefficients are set out in the formulae or tables, the method of use of which is set forth in **9.5.1.1** to **9.5.1.4**. Interpolation by proportion can be made, when necessary. Extrapolation shall never be made.

9.5.1.1 *Spacing* — Refer to Table 6.

9.5.1.2 *Flow coefficient* — The flow coefficient α is given by the following empirical formula, when D is expressed in millimetre:

$$\alpha = A + B\sqrt{\frac{10^6}{Re_D}}$$

where

$$A = 0{\cdot}5922 + 0{\cdot}4252\left[\frac{0{\cdot}3871}{D^2\beta^2 + 0{\cdot}254\,D} + \beta^4 + 1{\cdot}25\beta^{16}\right], \text{ and}$$

$$B = 0{\cdot}000025 + 0{\cdot}002325\,[\beta + 1{\cdot}75\,\beta^4 + 10\beta^{12} + 0{\cdot}07874\,D\beta^{16}]$$

NOTE 1 — Most of experiments from which the above values have been obtained were made in steel pipes in good condition for which an average value of $k = 0{\cdot}05$ mm was taken. The above formula should be used only when $\frac{D}{k} \geqslant 1000$ (k has the same meaning as in Notes 1 and 2 of Table 8).

NOTE 2 — If D is less than 50 mm or more than 760 mm this type of orifice plate shall not be used. Note also that β shall be between 0·1 and 0·8.

NOTE 3 — α can be computed by the above formula only if Re_D is within the limiting values given in Table 10.

9.5.1.3 *Limiting values of Reynolds number* Re_D — Table 10 shows as a function of β and D the lower and upper limits of Re_D between which α has been determined experimentally.

9.5.1.4 *Expansibility (expansion) factor* ϵ — The empirical formula for computing the expansibility (expansion) factor ϵ is as follows:

$$\epsilon = 1 - (0{\cdot}41 + 0{\cdot}35\beta^4)X$$

This formula is applicable only for value of β, D, $\frac{D}{k}$ and Re_D as stated in the Notes of **9.5.1.2**. Test results for determination of ϵ are known for air, steam and natural gas only. However, there is no known objection to using the same formula for other gases and vapours the isentropic exponent of which is known. Moreover it is applicable only if $\frac{P_2}{P_1} \geqslant 0{\cdot}75$.

9.6 Coefficient of Orifice Plates with Flange Taps

9.6.1 *Coefficients* — The coefficients are set out in the form of formulae or tables the method of use of which is set forth in the accompanying clause. Interpolation by proportion can be made when necessary. Extrapolation shall never be made.

9.6.1.1 The flow coefficient α is given by the following empirical formula when d, and D are expressed in millimetres. The flow coefficient

$$\alpha = \alpha' \left(1 + \frac{\beta A}{Re_D}\right)$$

where

$$\alpha' = \alpha_e \left[\frac{10^6 d}{10^6 d + 381\,A}\right]$$

$$\alpha_e = 0{\cdot}599\,3 + \frac{0{\cdot}177\,8}{D} + \left(0{\cdot}364 + \frac{0{\cdot}383\,0}{\sqrt{D}}\right)\beta^4$$

$$+ 0{\cdot}4\left(1{\cdot}6 - \frac{25{\cdot}40}{D}\right)^5 \left[\left(0{\cdot}07 + \frac{12{\cdot}70}{D}\right) - \beta\right]^{5/2}$$

$$- \left(0{\cdot}009 + \frac{0{\cdot}863\,6}{D}\right)(0{\cdot}5 - \beta)^{\frac{3}{2}} + \left(\frac{41\,935}{D^2} + 3\right)(\beta - 0{\cdot}7)^{\frac{5}{2}}$$

$$A = 0{\cdot}039\,37\ d\ (830 - 5\,000\ \beta + 9\,000\beta^2 - 4\ 200\beta^3 + \frac{2\,671}{\sqrt{D}}$$

NOTE 1 — In the above equation, when β has a value such that some of the terms of the equation are imaginary numbers, these terms should be taken to equal to zero.

NOTE 2 — Most of the experiments leading to the above values have been carried out in steel pipes in good condition for which the average value of k was taken as 0·05 mm. The above formula should be used only when $\frac{D}{k} \geqslant 1\,000$ (k here has the same meaning as in Notes 1 and 2 of Table 8).

NOTE 3 — If D is less than 50 mm or more than 760 mm, this type of orifice plate shall not be used. Also, the value of β shall be between 0·1 and 0·75.

NOTE 4 — α can be computed by the above formula only if Re_D remains between the limiting values given in Table 11.

9.6.1.2 *Limiting values of Reynolds number* Re_D — Table 11 gives, as a function of β and D, the lower and upper limiting values of Re_D between which α has been determined experimentally.

9.6.1.3 *Expansibility (expansion) factor* ϵ — The empirical formula for computing the expansibility (expansion) factor ϵ is given as $\epsilon = 1 - (0{\cdot}41 + 0{\cdot}35\beta^4)X$. This formula is applicable only for values of β, D, $\frac{D}{k}$ and Re_D as stated in the Notes of **9.6.1.1**. Test results for determination of ϵ are known for air, steam and natural gas only. However, there is no known objection to using the same formula for other gases and vapours the isentropic exponent of which is known. Moreover, it is applicable only if

$$\frac{P_2}{P_1} \geqslant 0{\cdot}75.$$

9.7 Pressure Loss $\Delta\tilde{\omega}$ — For the three standard orifice plates, the pressure loss $\Delta\tilde{\omega}$ is related to the differential pressure ΔP by the following approximate formula:

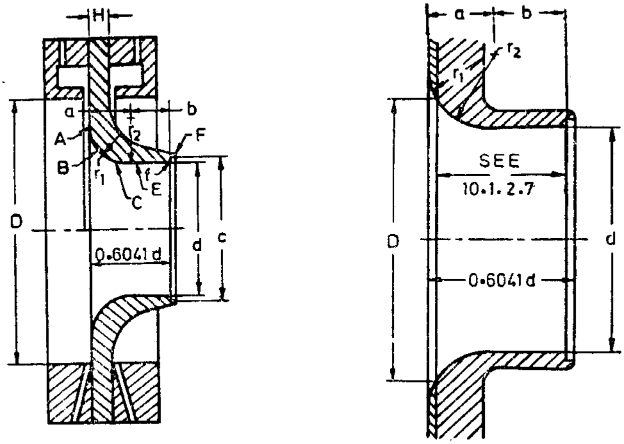
$$\Delta\tilde{\omega} \approx \left[\frac{1-\alpha\beta^2}{1+\alpha\beta^2}\right] \Delta P$$

10. NOZZLES

10.1 In this standard unless otherwise specified, a nozzle shall mean either of the following two types of standard nozzles:

ISA 1932 nozzle, and Long Radius nozzle
which are different and are described separately.

Figure 9 shows the section of an ISA 1932 nozzle by a plane passing through the centre-line of the throat. The letters in the text are those of the reference marks on the drawing.



$d < \frac{2}{3} D$

9A

$d > \frac{2}{3} D$

9B

FIG. 9 ISA 1932 NOZZLE

10.1.1 *General Shape* — The part of the nozzle inside the pipe is circular. The nozzle consists of a convergent portion, of rounded profile, and a cylindrical throat.

10.1.2 *Upstream Face*

10.1.2.1 This face may be characterized by describing:

a) a flat inlet part *A*, perpendicular to the centre-line;

b) a convergent section defined by two arcs of circumference, *B* and *C*;

c) a cylindrical throat E; and
d) a recess F.

10.1.2.2 The flat inlet part A is limited by a circumference centred on the axis of revolution, with a diameter of 1·5 d, and by the inside perimeter of the pipe, of a diameter D. When $d=2/3\ D$, the radial width of this flat part is zero. When d is more than 2/3 D, the upstream face of the nozzle does not include a flat inlet part within the pipe. In this case, the nozzle is manufactured as if D is greater than 3/2 d and the inlet flat part is then faced off so that its larger diameter is just equal to D (*see* Fig. 9B and **10.1.2.7**).

10.1.2.3 The arc of circumference B is tangential to the flat inlet part A, when $d<2/3$ D, while its radius r_1 is equal to $0{\cdot}2\ d \pm 10$ percent for $\beta<0{\cdot}5$; to $0{\cdot}2\ d \pm 3$ percent for $\beta \geqslant 0{\cdot}5$ and its centre is at 0·2 d from the inlet plane and at 0·75 d from the axial centre-line.

10.1.2.4 The arc of circumference C is tangential to arc of circumference B and to throat E. Its radius r_2 is equal to $d/3 \pm 10$ percent for $\beta<0{\cdot}5$ and to $d/3 \pm 3$ percent for $\beta \geqslant 0{\cdot}5$. Its centre is at $d/2+d/3=5/6\ d$ from the axial centre-line and at $a = \frac{12+\sqrt{39}}{60}\ d$ ($=0{\cdot}304\ 1\ d$) from the flat inlet part A.

10.1.2.5 The throat E has a diameter d and a length $b=0{\cdot}3\ d$. The value d of the diameter of the throat shall be taken as the mean of the measurements of at least four diameters situated in meridian planes distributed at approximately 45° angles to each other.

The throat shall be cylindrical. Any diameter of any cross-section shall not differ by more than 0·05 percent from the value of the mean diameter. This requirement is considered as satisfied when the deviations in the length of any of the measured diameters comply with the said requirement in respect of deviation from the mean. Attention is called to the fact that it is possible to check circularity of an orifice bore within the accuracy required without measuring the mean diameter of the orifice bore itself.

10.1.2.6 The recess F has a diameter c equal to at least 1·06 d and a length equal to or less than 0·03 d. The outlet edge f should be sharp.

10.1.2.7 The total length of the nozzle, excluding the recess, is 0·604 1 d, when d is less than 2/3 D and is shorter, due to the inlet profile, if d is more than 2/3 D. The values below summarize the total length of the nozzle, excluding the recess, as a function of β :

Values of β	*Total Length of the Nozzle, Excluding the Recess*
$0{\cdot}32 \leqslant \beta \leqslant \frac{2}{3}$	$0{\cdot}604\ 1\ d$
$\frac{2}{3} < \beta \leqslant 0{\cdot}8$	$\left[0{\cdot}404\ 1 + \left(\frac{0{\cdot}75}{\beta} - \frac{0{\cdot}25}{\beta^2} - 0{\cdot}522\ 5\right)^{\frac{1}{2}}\right] d$

10.1.2.8 The profile of the convergent inlet shall be checked by means of a template. Two diameters of the convergent inlet in the same plane perpendicular to the axial centre-line shall not differ one from the other by more than 0·1 percent of their mean value.

10.1.2.9 The surface of the upstream face shall be polished so as to keep the maximum roughness (peak-to-hollow height) under 0·000 3 *d*.

10.1.3 *Downstream Face*

10.1.3.1 Thickness *H* shall not exceed 0·1 *D*.

10.1.3.2 Apart from the above condition, the profile and surface finish of the downstream face are not specified (taking into account **10.1.1**).

10.1.4 *Material and Manufacture* — The details given in **9.1.9.1, 9.1.9.2** and **9.1.9.3** apply *mutatis mutandis* to the manufacture of ISA 1932 nozzle.

10.1.5 *Pressure Taps* — Corner pressure taps only shall be used with ISA 1932 nozzle. They shall comply with the requirements given in **9.2.5**.

10.1.6 *Installation of Nozzle*

10.1.6.1 The details given in **9.3** apply *mutatis mutandis* to the installation of ISA 1932 nozzle.

10.1.6.2 The measuring section of the pipe shall comply with conditions of **5.3** and **5.4**.

10.1.7 *Coefficients of ISA 1932 Nozzles* — The coefficients are given in the form of tables, the method of use of which is set down in the accompanying clause. Interpolation by proportion can be made, when necessary. Extrapolation shall never be made.

10.1.7.1 *Flow coefficient* α — The flow coefficient α is given by $\alpha = \alpha'' \times r_{ReD}$. Table 12 gives the values of α'' as a function of β^4 and Re_D, α'' is not dependent on D.

NOTE — This type of nozzle shall not be used when
$\beta^4 < 0{\cdot}01$ or $\beta^4 > 0{\cdot}41$ ($\beta < 0{\cdot}32$ or $\beta > 0{\cdot}8$)
$D < 50$ mm or $D > 1\,000$ mm
Re_D is outside the tabulated values.

10.1.7.2 *Correction factor for roughness* r_{Re} — The correction factor r_{Re} for pipe roughness is computed by the following formula:

$$r_{Re} = (r_0 - 1)\left(\frac{\log_{10} Re_D}{5{\cdot}5}\right)^2 + 1$$

with the exception that if $Re_D \geqslant 3{\cdot}2 \times 10^5$ then $r_{Re} = r_0$.

Table 13 gives values of r_0 as a function of β^2 and of D/k.

NOTE — Refer to **9.4.1.2** for computation of D/k.

10.1.7.3 *Expansibility (expansion) factor* ϵ — The expansibility (expansion) factor for nozzles is computed by the following formula :

$$\epsilon = \left[(1-x)^{2/\kappa} \left(\frac{\kappa}{\kappa-1}\right) \left\{ \frac{1-(1-x)^{\frac{\kappa-1}{\kappa}}}{x} \right\} \left\{ \frac{1-\beta^4}{1-\beta^4\,(1-x)^{2/\kappa}} \right\} \right]^{\frac{1}{2}}$$

This formula is applicable only for values of β, D, and Re_D as stated in the Note under **10.1.7.1**. Test results for determination of ϵ are known for air, steam and natural gas only. However, there is no known objection to using the same formula for other gases and vapours the isentropic exponent of which is known.

Moreover it is applicable only if $\frac{P_2}{P_1} \geqslant 0{\cdot}75$.

10.1.8 *Pressure Loss* $\Delta\tilde{\omega}$ — Refer to **9.7**.

10.2 Long-Radius Nozzles — There are two types of long-radius nozzles, which are called:

a) high-ratio nozzle ($0{\cdot}25 \leqslant \beta \leqslant 0{\cdot}8$), and
b) low-ratio nozzle ($0{\cdot}20 \leqslant \beta \leqslant 0{\cdot}5$).

For β values between 0·25 and 0·5, either design may be used. Figure 10 shows the sections of long-radius nozzles by planes passing through the orifice centre-lines. The letters in the text are those of the reference marks on the drawing. Both types of nozzles consist of a convergent inlet, whose shape is quarter ellipse and a cylindrical throat. That part of the nozzle which is inside the pipe shall be circular, with the possible exception of the holes of the pressure taps.

10.2.1 *Shape of High-Ratio Nozzle*

10.2.1.1 The upstream inner face may be characterized by:

a) a convergent section *A*,
b) a cylindrical throat *B*, and
c) a bevel or recess or plain end *C* (optional).

10.2.1.2 The convergent section *A* has the shape of a quarter ellipse. The centre of the ellipse is $1/2\ D$ from the axial centre-line. The major centre-line of the ellipse is parallel to the axial centre-line. The value of half the major centre-line is $1/2\ D$. The value of half the minor centre-line is $1/2\ (D-d)$. The profile of the convergent section shall be checked by means of a template. Two diameters of the convergent in the same plane perpendicular to the centre-line shall not differ one from the other by more than 0·1 percent of their mean value.

10.2.1.3 The throat *B* has a diameter *d* and a length $0{\cdot}6\ d$. The value *d* of the diameter of the throat shall be taken as the mean of the measurements of at least four diameters situated in meridian planes distributed at approximately 45° angles to each other. The throat shall be cylindrical.

Any diameter of any cross section shall not differ by more than 0·05 percent from the value of the mean diameter. This requirement is considered as satisfied when the deviations in the length of any of the measured diameters comply with the said requirement in respect of deviation from the mean.

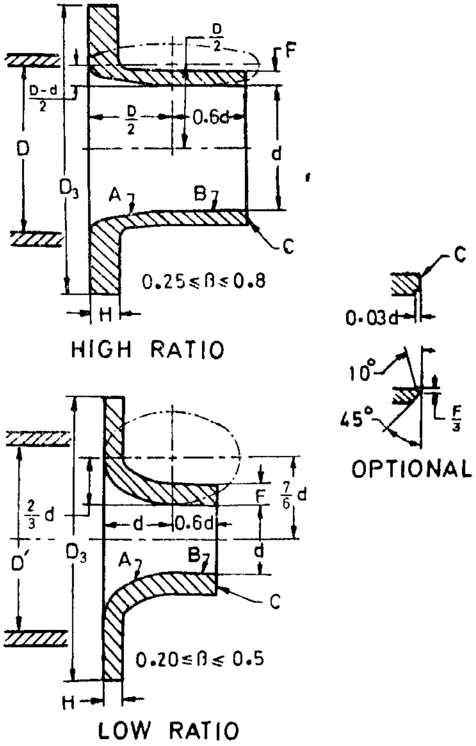


Fig. 10 Long-Radius Nozzles

10.2.1.4 The inside bevel shall have a slope of 10° and the bevel height is equal to the third of the thickness at the end of throat *F*. The outside bevel angle shall be 45°.

10.2.1.5 The recess *C* shall have a diameter of 1·06 *d* and a length of 0·03 *d*.

10.2.1.6 The distance between the pipe-wall and the outside face of the throat shall be greater than or equal to 3 mm.

10.2.1.7 The thickness *H* shall be greater than or equal to 3 mm and less than or equal to 0·15 *D*. The thickness *F* of the throat shall be between 3 and 13 mm.

10.2.1.8 The surface of the upstream face shall be polished so that the maximum roughness (peak-to-hollow height) is less than 0·0003 d.

10.2.1.9 The shape of the downstream (outside) face is not specified but shall comply with **10.2.1.6** and **10.2.1.7**. That part of the nozzle which is inside the pipe shall be circular, with the possible exception of the holes of the pressure taps.

10.2.2 *Shape of Low-Ratio Nozzle*

10.2.2.1 The requirements given in **10.2.1** for the high-ratio nozzle apply to the low-ratio nozzle, with the exception of the shape of the ellipse itself, which is given in **10.2.2.2**.

10.2.2.2 The convergent inlet A has the shape of a quarter ellipse. The centre of the ellipse is at a distance $\frac{d}{2}+\frac{2d}{3}=\frac{7}{6}d$ from the axial centre-line. The major centre-line of the ellipse is parallel to the axial centre-line. The value of half the major centre-line is d. The value of half the minor centre-line is $\frac{2}{3}d$.

10.2.3 *Material and Manufacture* — The requirement given in **9.1.9.1**, **9.1.9.2** and **9.1.9.3** shall apply *mutatis mutandis* to the manufacture of long radius nozzles.

10.2.4 *Pressure Taps*

10.2.4.1 The pressure taps shall comply with the descriptions in **9.2.2** and **9.2.3**.

10.2.4.2 The position of the upstream tap shall be at $1D\,^{+0{\cdot}2D}_{-0{\cdot}1D}$ from the inlet face of the nozzle. The centre-line of the downstream tap shall be at $0{\cdot}50\ D \pm 0{\cdot}01D$ from the inlet face of the nozzle with the exception that it shall not in any case be further downstream than the nozzle outlet.

10.2.4.3 The upstream and downstream pressure taps breakthrough the inside wall of the pipe.

10.2.5 *Installation of Nozzle*

10.2.5.1 The requirements given in **9.3** apply *mutatis mutandis* to the installation of long radius nozzles.

10.2.5.2 The measuring portion of the pipe shall comply with conditions in **5.3.2** to **5.3.5**, **5.3.9** to **5.3.14** and **5.4**.

10.2.6 *Coefficients of Long-Radius Nozzles* — The coefficients are given in a table, the method of use of which is set down in the accompanying clause. Interpolation by proportion can be made, when necessary. Extrapolation shall never be made.

10.2.6.1 *Coefficient of discharge C* — The coefficients are same for both types of long-radius nozzles when the taps are in accordance with **10.2.4**.

Table 14 gives the coefficient of discharge C as a function of β and Re_D (C is defined in **6.4.6**).

NOTE 1 — D shall be between 50 and 200 mm.

NOTE 2 — Within these limits, C is independent of the value of D.

NOTE 3 — Table 14 shall not be extrapolated either with respect to β or to the upper and lower limiting values of Re_D.

NOTE 4 — The coefficients of Table 14 shall be used only when D/k (as defined in Table 8) is $\geqslant$ 1 000.

10.2.6.2 *Expansibility (expansion) factor* ϵ — The expansibility factor ϵ for long-radius nozzles is computed from the following formula:

$$\epsilon = \left[(1-x)^{2/\kappa} \left(\frac{\kappa}{\kappa-1}\right) \left(\frac{1-(1-x)^{\frac{\kappa-1}{\kappa}}}{x}\right) \left(\frac{1-\beta^4}{1-\beta^4(1-x)^{2/\kappa}}\right) \right]^{\frac{1}{2}}$$

This formula is applicable only for values of β, D, and Re_D, as stated in the note under **10.2.6.1**. Test results for determination of ϵ are known for air, steam and natural gas only. However, there is no known objection to using the same formula for other gases and vapours the isentropic exponent of which is known.

Moreover it is applicable only if $\frac{P_2}{P_1} \geqslant 0{\cdot}75$.

11. CLASSICAL VENTURI TUBES

11.1 Field of Application — Classical venturi tubes cause a much smaller pressure loss than standard orifice plates or nozzles with the same area ratio. Their range of use, within the compass of this standard, depends on the way they are manufactured, as indicated in **11.1.1**, **11.1.2** and **11.1.3**.

11.1.1 Classical venturi tubes with a rough-cast convergent may be used in pipes in the range of diameters between 100 and 800 mm provided the diameter ratio β lies between 0·3 and 0·75 inclusive ($0{\cdot}09 \leqslant \beta^2 \leqslant 0{\cdot}56$).

11.1.2 Classical venturi tubes with a machined convergent may be used in pipes in the range of diameters between 50 and 250 mm provided the diameter ratio β lies between 0·4 and 0·75 inclusive ($0{\cdot}16 \leqslant \beta^2 \leqslant 0{\cdot}56$).

11.1.3 Classical venturi tubes with a rough welded sheet-iron convergent may be used in pipes in the range of diameters between 200 and 1200 mm provided the diameter ratio β lies between 0·4 and 0·7 inclusive ($0{\cdot}16 \leqslant \beta^2 \leqslant 0{\cdot}49$).

11.2 Geometric Profile — Figure 11 represents a section passing through the centre-line of the throat of a classical venturi tube. The letters are for reference purposes. The classical venturi tube is made up of an entrance cylinder A connected to a conical convergent B, a cylindrical throat C and a conical divergent E. The internal surface of the device shall be a surface of revolution and concentric with the pipe centre-line. The convergent and the cylindrical throat are considered as coaxial by mere visual inspection.

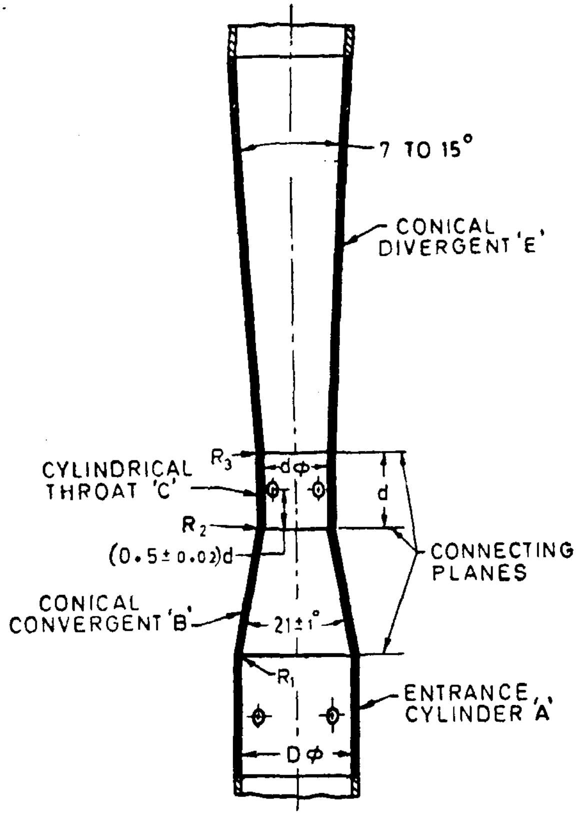


FIG. 11 GEOMETRIC PROFILE OF THE CLASSICAL VENTURI TUBE

11.2.1 The entrance cylinder *A* shall have a diameter *D* which shall not differ from the pipe inside diameter by more than 0·01 *D*. The minimum cylinder length, measured from the plane containing the intersection of cone frustum *B* with cylinder *A*, may vary as a result of the manufacturing process (*see* **11.2.7.1**, **11.2.8.1** and **11.2.9.1**). It is however recommended that it is equal to 1 *D* whenever possible. The entrance cylinder diameter *D* shall be measured very carefully in the plane of the upstream pressure taps. The number of measurements shall be equal to that of the pressure taps (with a minimum of four). The diameters shall be measured near each pair of pressure taps, and also between these pairs. The mean value of all these measurements shall be taken as the value of *D* in the calculation. Diameters shall also be measured in planes other than the plane of the pressure taps. No diameter along the entrance cylinder shall differ by more than

0·4 percent from the value of the mean diameter. This requirement is satisfied when the difference in the length of any of the measured diameters complies with the said requirement in respect of the mean of the measured diameters.

11.2.2 The convergent B shall be conical and shall have an included angle of $21 \pm 1°$ for any type of classical venturi tube. It is delimited upstream by the plane containing the intersection of cone frustum B with the entrance cylinder A (or their prolongations) and downstream by the plane containing the intersection of cone frustum B with the throat C (or their prolongations).

The overall length of the convergent B measured parallel to the revolution centre-line is therefore approximately equal to $2{\cdot}7\ (D\text{-}d)$. The convergent B is blended to the entrance cylinder A by a curvature of radius R_1, the value of which depends on the type of classical venturi tube.

The profile of the convergent section shall be checked by means of a template. The deviation between the template and the conical section of the convergent shall not exceed, in any place, 0·4 percent of D. The internal surface of the conical section of the convergent is taken as being a surface of revolution when two diameters situated in the same plane perpendicular to the revolution centre-line do not differ from the value of the mean diameter by more than 0·4 percent. It shall be checked in the same way that the joining curvature with a radius R_1 is a surface of revolution.

11.2.3 The throat C shall be cylindrical with a diameter d. It is delimited upstream by the plane containing the intersection of cone frustum B with the throat C (or their prolongations) and downstream by the plane containing the intersection of the throat C with the cone frustum E (or their prolongations). The length of the throat C, that is the distance between those two planes, shall be equal to d whatever the type of the classical venturi tube. The throat C is connected to the convergent B by a curvature of radius R_2 and to the divergent E by a curvature of radius R_3. The values of R_2 and R_3 depend on the type of classical venturi tube. The diameter d shall be measured very carefully in the plane of the throat pressure taps. The number of measurements shall be equal to that of the pressure taps (with a minimum of four). The diameters shall be measured near each pair of pressure taps and also between these pairs. The mean value of all these measurements shall be taken as the value of d in the calculations. Diameters shall also be measured in planes other than the plane of the pressure taps.

No diameter along the throat shall differ by more than 0·1 percent of the value of the mean diameter. This requirement is satisfied when the difference in the length of any of the measured diameters complies with the said requirement in respect of the mean of the measured diameters.

The throat of the venturi tube shall be machined or be of equivalent smoothness over the whole of its length. It shall be checked that the joining curvatures into the throat with radii R_2 and R_3 are surfaces of revolution as described in **11.2.2**. This requirement is satisfied when diameters, situated in the same plane perpendicular to the revolution centre-line, do not differ from the value of the mean diameter by more than 0·1 percent. The values of the radii of curvature R_2 and R_3 shall be checked by means of a template. The deviation between the template and the venturi tube shall evolve in a regular way for each curvature so that the single maximum deviation that is measured occurs at approximately mid-way along the template profile. The value of this maximum deviation shall not exceed 0·02 d.

11.2.4 The divergent E shall be conical and shall have an included angle lying between 7° and 15°; it is however recommended that an angle between 7° and 8° be chosen.

11.2.5 A classical venturi tube is called 'truncated' when the outlet diameter of the divergent is less than the diameter D and 'not truncated' when the outlet diameter is equal to diameter D. The divergent can be truncated down by about 35 percent of its length without greatly modifying the pressure loss in the device. This allows a reduction in the divergent length without the angle being modified.

11.2.6 The roughness criterion R_a of the throat and that of the adjacent curvatures shall be less than $10^{-5}\,d$ and if possible equal to $0{\cdot}5 \times 10^{-5}\,d$. The divergent is rough-cast. Its internal surface shall be clean and smooth. The roughness of other parts of the classical venturi tube depends on the type considered.

11.2.7 *Characteristics of the Profile of the Classical Venturi Tube with a Rough-Cast Convergent*

11.2.7.1 The minimum length of entrance cylinder A shall be equal to the smaller of the two quantities 1D and 0·25 D+250 mm (*see* **11.2.1**).

11.2.7.2 The radius of curvature R_1 shall be equal to 1·375 D (±20 percent).

11.2.7.3 The radius of curvature R_2 shall be equal to 3·625 d (±0·125 d).

11.2.7.4 The length of the cylindrical part of the throat shall be not less than $\frac{1}{3}\,d$. In addition the length of the cylindrical part lying between the end of the joining curvature R_2 and the plane of the pressure taps shall be not less than $\frac{1}{6}\,d$ (*see also* **11.2.3** for the throat length).

11.2.7.5 The radius of curvature R_3 shall lie between 5 and 15 d. Its value shall increase as the divergent angle decreases. A value close to 10 d is recommended for a divergent included angle close to 7°. Whatever may be the radius R_3 adopted, the length of the throat cylindrical part lying between the plane of the pressure taps and the beginning of the curvature of radius R_3 shall not be less than $\frac{1}{6}\,d$. (*see also* **11.2.3** for the throat length).

11.2.7.6 The internal surface of the convergent is sand-cast and rough-cast. It shall be smooth, free from cracks, fissures, depressions, irregularities and impurities. The roughness criterion R_a shall be less than $10^{-4}\,D$.

The internal surface of entrance cylinder *A* may also remain rough-cast if it shows the same surface finish as that of the convergent *B* and provided that it complies with **11.2.1**.

11.2.8 *Characteristics of the Profile of the Classical Venturi Tube with a Machined Convergent*

11.2.8.1 The minimum length of cylinder *A* shall be equal to 1 *D*.

11.2.8.2 The radius of curvature R_1 shall be less than 0·25 *D* and preferably equal to zero.

11.2.8.3 The radius of curvature R_2 shall be less than 0·25 *d* and preferably equal to zero.

11.2.8.4 The length of the throat cylindrical part lying between the end of the curvature R_2 and the plane of the pressure taps shall be not less than 0·25 *d*.

11.2.8.5 The radius of curvature R_3 shall be less than 0·25 *d* and preferably equal to zero. The length of the throat cylindrical part lying between the end of the curvature R_3 and the plane of the pressure taps shall be not less than 0·3 *d*.

11.2.8.6 The entrance cylinder and the covergent shall have a surface finish equal to that of the throat.

11.2.9 *Characteristics of the Profile of the Classical Venturi Tube with a Rough Welded Sheet-Iron Convergent*

11.2.9.1 The minimum length of entrance cylinder *A* shall be equal to 1*D*.

11.2.9.2 There shall be no joining curvature between entrance cylinder *A* and convergent *B* other than that resulting from the welding.

11.2.9.3 There shall be no joining curvature between convergent *B* and throat *C* other than that resulting from the welding.

11.2.9.4 There shall be no joining curvature between the throat and the divergent.

11.2.9.5 The internal surface of entrance cylinder *A* and convergent *B* shall be clean and free from encrustation and welding deposits. It may be galvanized. Its roughness criterion R_a shall be about $5 \times 10^{-4} D$. The height of the internal welded seams shall not exceed 5 percent of the diameter at the plane of the weld.

11.3 Material and Manufacture

11.3.1 The classical venturi tube may be manufactured of any material and in any way, provided it is in accordance with the foregoing description and will remain so during use.

11.3.2 It is recommended that convergent *B* and throat *C* be joined as one part. It is recommended that in the case of a classical venturi tube with a machined convergent, the throat and the convergent are manufactured from one piece of material. If however they are made in two separate parts they shall be assembled before the internal surface is finally machined.

11.3.3 Particular care shall be given to the centring of the divergent *E* on the throat. Thus there shall be no step in diameters between the two parts. This can be established by touch before the classical venturi tube is installed, but after the divergent has been assembled with the throat section.

11.3.4 When a lining is added in the throat, it shall be machined after being installed inside the classical venturi tube.

11.4 Pressure Taps

11.4.1 The upstream and throat pressure taps shall be made in the form of separate pipe-wall pressure taps interconnected by annular chambers. The diameters of these taps shall lie between 4 and 10 mm and moreover shall never be greater than $0{\cdot}1\ D$ for the upstream taps and $0{\cdot}13\ d$ for the throat pressure taps. It is recommended that pressure taps as small as compatible with the fluid are used. The annular chamber shall be provided with at least four pressure taps for upstream and throat pressure measurements. The centre-lines of the pressure taps shall form equal angles with each other and shall be contained in a plane perpendicular to the centre-line of the classical venturi tube. At the point of breakthrough, the hole of the pressure tap shall be circular. The edges should be flush with the pipe-wall, free from burrs and generally have no peculiarities and if joining curvatures are required the radius shall not exceed one-tenth of the diameter of the pressure tap. It is recommended that the pressure pipes to the secondary devices shall have a length at least equal to twice the diameter of the pressure taps. Conformity of the pressure taps with the two foregoing requirements can be assessed by mere visual inspection. Except in special cases, there shall be an even number of pressure taps.

11.4.2 The spacing of a pressure tap is the distance, measured on a straight line parallel to the centre-line of the classical venturi tube, between the centre-line of the pressure tap and the reference plane defined below.

For the classical venturi tube with a rough-cast convergent, the spacing between the upstream pressure taps situated on the entrance cylinder and the intersection plane between the prolongations of convergent *B* and entrance cylinder *A* shall be $0{\cdot}5\ D \pm 0{\cdot}25\ D$ for D between 100 and 150 mm and $0{\cdot}5D\ {}^{+0}_{-0{\cdot}25D}$ for D between 150 and 800 mm.

For classical venturi tubes with a machined convergent and with a rough welded sheet-iron convergent, the spacing between the upstream pressure taps and the intersection plane of entrance cylinder *A* and convergent *B* (or their prolongations) shall be $0{\cdot}5\ D \pm 0{\cdot}05\ D$.

For all types of classical venturi tubes, the spacing between the throat pressure taps and the intersection plane of convergent B and throat C (or their prolongations) shall be $0{\cdot}5\ d \pm 0{\cdot}02\ d$.

11.4.3 The area of the free cross section of the annular chamber of the pressure taps shall be equal to or more than half the total area of the tapping holes connecting the chamber to the pipe.

It is recommended that the chamber section given above should be doubled when the classical venturi tube is used with a minimum upstream straight length from a fitting causing non-symmetrical flows.

11.5 Installation of the Classical Venturi Tube

11.5.1 *Upstream Pipe* — In order to use the discharge coefficients given in **11.6**, the upstream pipe shall be smooth from the upstream fitting defined in Table 2 or from a length at least equal to 15 D if the pipe is straight over a length exceeding this value. In every case the roughness criterion R_a of the upstream pipe shall be less than $10^{-4}D$ on a length at least equal to 2 D measured upstream from the classical venturi tube.

11.5.2 *Position of Line* — The details given in **9.3.1** and **9.3.1.1** apply *mutatis mutandis* to the installation of classical venturi tubes.

11.5.3 *Installation* — The measuring pipe shall satisfy the general requirements for valid measurements as indicated in **5.3** and **5.5**.

11.6 Discharge Coefficients *C*

11.6.1 *Limits of Use* — Whatever may be the type of the classical venturi tube, a simultaneous use of extreme values for D, β and Re_D shall be avoided (*see* **5.1.2**). For installation outside the limits defined in **11.6.2, 11.6.3** and **11.6.4** for D, β or Re_D, it remains necessary to calibrate separately the primary element in its actual conditions of service. The effects of Re_D, $\frac{D}{k}$ and β or C are not yet sufficiently known for it to be possible to give reliable values of C outside the limits defined for each type of classical venturi tube.

11.6.2 *Discharge Coefficient of the Classical Venturi Tube with a Rough-Cast Convergent* — The value of the discharge coefficient C is:

$$C = 0{\cdot}984$$

within the following limits:

$$\begin{aligned} 100 \text{ mm} &\leqslant D \leqslant 800 \text{ mm} \\ 0{\cdot}3 &\leqslant \beta \leqslant 0{\cdot}75 \\ 2 \times 10^5 &\leqslant Re_D \leqslant 2 \times 10^6 \end{aligned}$$

11.6.3 *Discharge Coefficient of the Classical Venturi Tube with a Machined Convergent* — The value of the discharge coefficient C is:

$$C = 0{\cdot}995$$

within the following limits:

$$\begin{aligned} 50 \text{ mm} &\leqslant D \leqslant 250 \text{ mm} \\ 0{\cdot}4 &\leqslant \beta \leqslant 0{\cdot}75 \\ 2 \times 10^5 &\leqslant Re_D \leqslant 1 \times 10^6 \end{aligned}$$

11.6.4 *Discharge Coefficient of the Classical Venturi Tube with a Rough-Welded Sheet-Iron Convergent* — The value of the discharge coefficient C is:

$$C = 0{\cdot}985$$

within the following limits:

$$\begin{aligned} 200\ \text{mm} &\leqslant D \leqslant 1200\ \text{mm} \\ 0{\cdot}4 &\leqslant \beta \leqslant 0{\cdot}7 \\ 2 \times 10^5 &\leqslant Re_D \leqslant 2 \times 10^6 \end{aligned}$$

11.7 Values of Expansibility (Expansion) Factor — The theoretical equation for the expansibility factor for venturi tubes is as follows:

$$\epsilon = \left[\left(\frac{\kappa \tau^{1/\kappa}}{\kappa - 1} \right) \left(\frac{1 - \beta^4}{1 - \beta^4 \tau^{2/\kappa}} \right) \left(\frac{1 - \tau^{\frac{\kappa - 1}{\kappa}}}{1 - \tau} \right) \right]^{1/2}$$

Values of the expansibility factor can be read from Table 15 for a range of isentropic exponents, pressure ratios and area ratio values. Interpolation by proportion can be made when necessary. Extrapolation shall never be made. In particular, it should be noted that there may be increasing differences between the actual and the theoretical conditions when the pressure ratio decreases below 0·75.

12. VENTURI NOZZLE

12.1 Field of Application → Standard venturi nozzles may cause a much smaller pressure loss than standard nozzles or orifice plates of the same area ratio. They may be used for measurement of gases flowing in pipes with diameter D varying from 65 to 500 mm and for area ratios β^2 varying from 0·1 to 0·6. The lower limit of D is dependent on the area ratio β^2 (*see* **12.6.1**) and the diameter of the throat pressure taps given in **12.4.3**.

Insufficient test data are available to confirm the discharge coefficients for diameter D greater than 500 mm but, as a guide, there is no reason to suppose that the coefficients will alter significantly from those obtained at a diameter of 500 mm.

12.2 Geometric Profile

12.2.1 The profile of a venturi nozzle (*see* Fig. 12 and 13) is axisymmetric. It consists of a convergent portion, with a rounded profile, a cylindrical throat and a divergent.

12.2.2 The upstream portion shall be identical with an ISA 1932 nozzle (as defined in **10.1.2**) as shown in Fig. 12 and 13 and described in the following clauses.

12.2.3 The flat inlet part A is limited by a circumference, centred on the axial centre-line, with a diameter of 1·5 d, and by the inside perimeter of the pipe, of diameter D. When d is equal to $\frac{2}{3}$ D, the radial width of this

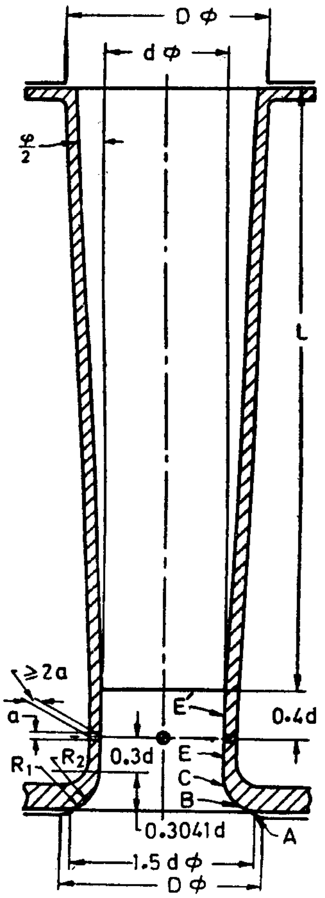


FIG. 12 VENTURI NOZZLE (TRUNCATED) — DIVERGENT $d < \frac{2}{3} D$

flat part is zero. When d is more than $\frac{2}{3} D$ the upstream face of the nozzle does not include a flat inlet part within the pipe. In this case, the nozzle is manufactured as if D were greater than $1 \cdot 5\, d$ and the inlet flat part is then faced off so that its largest diameter is just equal to D (*see* Fig. 13).

12.2.4 The arc of circumference B is tangential to the flat inlet part A, when $d < \frac{2}{3} D$. Its radius R_1 is equal to $0 \cdot 2\, d \pm 10$ percent for $\beta < 0 \cdot 5$, and to $0 \cdot 2\, d \pm 3$ percent for $\beta \geqslant 0 \cdot 5$. Its centre is at $0 \cdot 2\, d$ from the inlet plane and at $0 \cdot 75\, d$ from the axial centre-line.

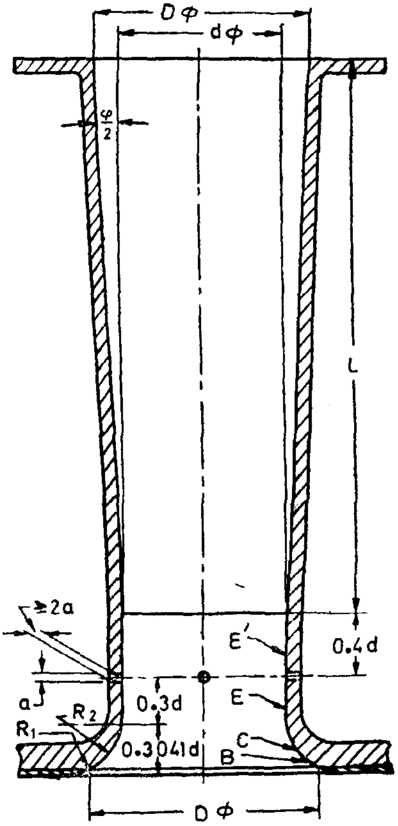


Fig. 13 Venturi Nozzle (Non-truncated Divergent $d > \frac{2}{3} D$)

12.2.5 The arc of *circumference C* is tangential to arc of circumference *B* and to throat *E*. Its radius R_2 is equal to $\frac{1}{3}\, d \pm 10$ percent for $\beta < 0{\cdot}5$ and to $\frac{1}{3} d \pm 3$ percent for $\beta \geqslant 0{\cdot}5$. Its centre is at $1/2\ d + \frac{5}{6}\ d$ from the axial centre-line and at $\frac{12 + \sqrt{39}}{60}\, d\ (= 0{\cdot}304\ 1\ d)$ from the flat inlet part *A*.

12.2.6 The throat *E* has a diameter *d* lying between 50 and approximately 390 mm and length $b = 0{\cdot}3\ d$. The value *d* of the diameter of the throat shall be taken as the mean of the measurements of at least four diameters situated in meridian planes distributed at approximately 45° angles to each other. The throat shall be cylindrical. Any diameter of any cross section shall not differ by more than 0·1 percent from the value of the mean diameter. This requirement is considered as satisfied when the deviations in the length of any of the measured diameters comply with the said requirement in respect of deviation from the mean.

12.2.7 Part E' of the cylindrical throat shall have a length of 0·4 d (*see* Fig. 12 and 13). For the specification of diameter d and the machining of the inside surface (*see* **12.2.6** and **12.2.10**).

12.2.8 The divergent (*see* Fig. 12 and 13) shall be connected with part E' of the throat without a rounded part, but any burrs shall be removed. The included angle of the divergent ϕ shall be $\leqslant$ 30°. The length L of the divergent has practically no influence on the flow coefficient α. The included angle of the divergent, and hence the length, does influence, however, the pressure loss.

12.2.9 The venturi nozzle may be truncated in the same way as the classical venturi tube (*see* **11.2.5**).

12.2.10 The internal surfaces of the venturi nozzles shall have a roughness criterion $R_a < 10^{-5}\ d$ (*see* **5.1.2**).

12.3 Material and Manufacture

12.3.1 The venturi nozzle may be manufactured of any material, provided it is in accordance with the foregoing description (**12.2.1** to **12.2.10**), and will remain so during use. In particular, the venturi nozzle shall be clean when the measurements are made.

12.3.2 The venturi nozzle is usually made of metal, and preferably shall be erosion proof and corrosion proof.

12.4 Pressure Taps

12.4.1 *Angular Position of Pressure Taps*

12.4.1.1 The centre-line of the pressure taps may be located in any meridian half-planes of the pipe.

12.4.1.2 The centre-line of the wall-tap in the throat piezometer ring and that of the upstream tap may be located in different meridian half-planes, whether the upstream tap be a single tap or a wall-tap in the upstream piezometric ring. However, attention is drawn to the fact that in any case the differential pressure as computed from indications collected from these pressure taps shall be in accordance with the definition of **6.2.3**, especially as regards gravitational energy variations. Moreover, it is preferable that these pressure taps shall lie in the same meridian half-plane when measuring fluids where a temperature difference might occur in the pressure pipes connecting the taps to the secondary device, since the pressure lines can run together and be lagged more simply than with any other configuration.

12.4.2 Upstream pressure taps shall be corner taps only as defined in the following clauses.

12.4.2.1 They shall comply with the requirements given in **9.2.5.1** to **9.2.5.4**. For venturi nozzle refer to Fig. 14 instead of Fig. 8 which are for orifice plates.

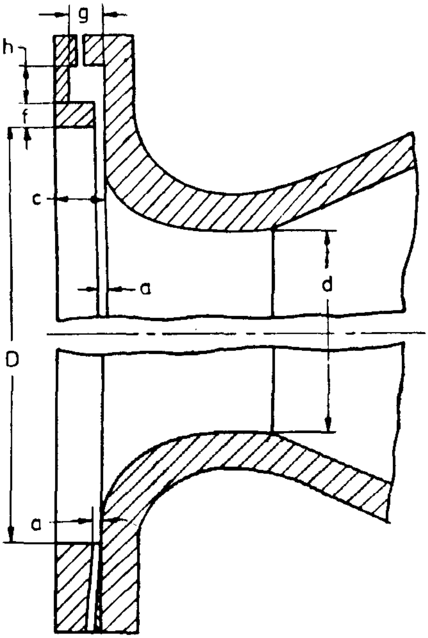


FIG. 14 PIEZOMETER RINGS AND CORNER TAPS (INDIVIDUAL TAPS)

12.4.2.2 If single pressure taps of the type shown in Fig. 14 are used, then the centre-line of each tap shall cut the centre-line of the pipe and make with it an angle of 90 ± 5°. If there are several single pressure taps in the same plane, then their centre-line shall form equal angles with each other. The diameters of single pressure taps are given in **9.2.5.3**.

12.4.3 *Throat Pressure Taps*

12.4.3.1 Throat pressure taps shall be single pressure taps leading into an annular chamber. Annular slits or interrupted slits shall not be used.

12.4.3.2 There shall be at least four single pressure taps; their centre-lines shall make equal angles one to another, and lie in the plane perpendicular to the centre-line of the venturi nozzle, which is the imagined border between the parts E and E' of the cylindrical throat.

12.4.3.3 The taps shall always be wide enough to ensure that clogging by dirt plugs or gas bubbles is prevented.

12.4.4 *Piezometer Rings*

12.4.4.1 The inner diameter d' of the upstream piezometer ring shall be between D and 1·02 D, and the axial length shall be smaller than 0·2 D. The diameter ratio β shall be calculated using the pipe diameter D and not the diameter d' of the ring.

12.4.4.2 The thickness f of the annular slit (or the length of single taps) shall be equal to or more than twice the width a of the slit (or of the diameter a of the single taps). The area of the free cross section of the annular chamber $g \times h$ shall be equal to or more than half the total area of the tap holes connecting the chamber to the pipe.

12.4.4.3 All surfaces of the piezometer ring which will be in contact with the measured fluid shall be clean and have a good machine finish.

12.4.4.4 The pressure taps connecting the annular chambers to the secondary devices shall be wall pressure taps, circular at the point of breakthrough and with diameters between 4 and 10 mm.

12.5 Installation of the Venturi Nozzle

12.5.1 *Position of Line*—The details given in **9.3.1** and **9.3.1.1** apply *mutatis mutandis* to the installation of venturi nozzle.

12.5.1.1 *Pipe diameter* — The value D of the diameter of the pipe shall be taken as the mean of the measurements of several diameters situated in meridian planes at approximately equal angles to each other, and in different cross sections of a length equal to 2 D measured upstream from the upstream face of the primary element. At least four diameters shall be measured.

12.5.2 *Location of Venturi Nozzle and Rings*

12.5.2.1 The venturi nozzle shall be placed in the pipe in such a way that the fluid flows from the convergent part (nozzle) towards the divergent.

12.5.2.2 The centring of the venturi nozzle in the pipe or in the upstream piezometer ring is important. The shortest distance between the centre-line of the venturi nozzle and the centre-lines of the pipe on the upstream and downstream sides shall be less than 0·015 D $(\beta - 1)$. The upstream piezometer ring (the inner diameter of which should be between D and 1·02 D) shall be centred so that it does not protrude into the pipe at any point.

12.5.3 *Fixing and Gaskets*

12.5.3.1 The method of fixing and tightening shall be such that once the venturi nozzle has been finally fixed in the proper position, it remains so.

12.5.3.2 Gaskets, if used, shall be made and inserted in such a way that they do not protrude at any point inside the pipe. They shall be as thin as possible.

12.5.3.3 The gaskets, if used, between the venturi nozzle and the annular chamber rings, shall not protrude inside the annular chambers.

12.5.4 *Static Pressure Measurement and Temperature Measurement*

12.5.4.1 The static pressure of the fluid shall be measured in the plane of the upstream pressure tap, by means of a wall pressure tap, as described in **12.4**. This tap shall be separate from the tap provided for measuring the upstream component of the differential pressure, unless the intention is to measure pressures separately at the upstream and throat pressure taps. If there is an upstream annular chamber, the static pressure shall be measured at the wall of this annular chamber.

12.5.4.2 The temperature of the fluid shall preferably be measured downstream of the primary element. The thermometer pocket shall take up as little space as possible. The distance between it and the venturi nozzle shall be at least equal to 5 D if the pocket is located downstream. When the pocket is located upstream, it shall lie more than 20 D from the venturi nozzle if its diameter is between 0·03 D and 0·13 D; if its diameter is less than or equal to 0·03 D, it shall lie more than 5 D from the venturi nozzle.

12.5.4.3 Any method of determining the static pressure and the temperature of the fluid is acceptable if it enables a reliable value of the temperature, the viscosity and the mass density of the fluid upstream of the element to be obtained without disturbing the flow measurement in any way.

12.5.5 *Thermal Insulation*

12.5.5.1 The measuring section and the pipe flanges shall be lagged over at least the whole length of the required straight runs.

12.5.5.2 It is however unnecessary to lag the pipe when the temperature of the fluid, between the inlet of the minimum straight length of the upstream pipe and the outlet of the minimum straight length of the downstream pipe, does not exceed any limiting value selected by the user as being sufficient for the accuracy of flow measurement which he requires.

12.6 Flow Coefficient α

12.6.1 The flow coefficient α of venturi nozzles in smooth pipe is dependent on the area ratio β^2 (*see* Table 16). In Table 16, β^4 instead of β^2 has been chosen as the parameter because it permits a linear interpolation of α. Extrapolation shall never be made. The flow coefficient α is given as $\alpha = \alpha_0 \times r_0$ where α_0 is the flow coefficient given in Table 16 and r_0 is the correction factor for pipe roughness (*see* **12.6.2**).

NOTE — This type of venturi nozzle shall not be used when
$\beta^4 < 0.01$ or $\beta^4 > 0.36$ ($\beta < 0.32$ or $\beta > 0.77$)
$D < 65$ mm or $D > 500$ mm
Re_D is outside the limits given in Table 18.
Furthermore, it is recalled that d shall lie between 50 mm and approximately 390 mm ($500\sqrt{0.6}$) (*see* **12.2.6**).

12.6.2 The values r_0 for venturi nozzles are same as ISA 1932 nozzles and Table 13 may be used for both. Guide values of the mean roughness of pipe walls, *k*, in millimetres are given in Table 17. For venturi nozzle, there is no need to correct r_0 for the influence of Reynolds number — usual with ISA 1932 nozzles — because within the limits of Re_D given in **12.6.3** the deviations of the corrected value r_{Re} from r_0 do not exceed + 0·2 percent and thus remain within the uncertainty with which r_0 is known.

12.6.3 Figure 15 shows the lower and upper limits (Re_D min and Re_D max) of Reynolds number for which the flow coefficient may be taken as constant with good approximation. These values are reproduced in Table 18 (as a function of β^4) which is of more practical use. Linear interpolation of Re_D is permissible on the basis of β^4; extrapolation shall never be made. The maximum values limit the range in which α_0 has been determined experimentally.

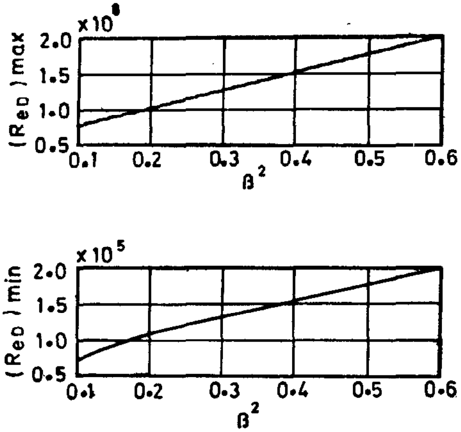


FIG. 15 STANDARD VENTURI NOZZLE-LIMITING VALUES OF REYNOLDS NUMBER Re_D FOR THE STANDARDIZED FLOW COEFFICIENT

12.7 Expansibility (expansion) factor for venturi nozzles may be determined in a manner similar to venturi tubes and is described in **11.7**.

13. FLOW MEASUREMENT BASED ON CALORIFIC VALUE AND COMPONENT ANALYSIS

13.1 Flow Measurement Based on Calorific Value — When the end use of the gas is a fuel, the quantity of the gas is frequently measured in terms of its calorific value and is related to the mass flow rate by the following equation:

$$q_H = q_m \times H_c$$

where

q_H = the flow rate based on calorific value and is expressed in kcal per second,

q_m = the mass flow rate of the gas in kg per second, and

H_c = the calorific value per kg of the gas flowing in kcal per kg.

13.2 Flow Measurement Based on Component Analysis — This method is used when the quantitative flow rate of a particular component present in a gas mixture is important as in the case of component sale or process feed stock. This is related to the mass flow rate as

$$q_A = q_m \times N_A$$

where

q_A = the mass flow rate of a particular component '*A*' present in the gas mixture expressed in kg per second,

q_m = the mass flow rate of the gas mixture in kg per second, and

N_A = the mass of component '*A*' present in unit mass of the gas mixture and is expressed in kg of component '*A*' per kg of the gas mixture.

14. SAMPLING

14.0 Sampling is a unit operation and is the most useful tool to obtain and interpret information pertaining to the composition and properties of gases. Important variations of conditions in different situations make it necessary to obtain a kind of sample as required by the kind of flow and the purpose of the work. For intermittent or batch flow the time of sampling is a matter of prime importance whereas for continuous flow the point of sampling is considered more important than the sampling time.

14.1 For continuous flow, sampling of the gas shall be carried out at least once a day for the determination of density. If the flow rate is based on component analysis then the composition of the gas shall also be tested, at least once a day, otherwise, it shall be tested at least once a month. For an interrupted or batch flow, samples shall be collected and tested at least once in every batch or once in a day, whichever is more frequent. Sampling and testing frequency specified in this clause is not mandatory, and may be decided upon by agreement between the buyer and the seller. Samples shall always be collected when the flow conditions are stable.

14.2 The sample shall be of sufficient volume to permit an accuracy of testing up to the limit of accuracy of the methods of testing, as may be required for the object in view. The number of tests to be made and the volume required for each test will determine the amount of sample. Normally 1 litre of sample is sufficient to test the molecular mass, density and composition of the gas.

14.3 Sampling point shall be located on the downstream side of the primary element and shall be at least 15 times of the pipe diameter away from the upstream face of the primary element.

14.4 The method of sampling shall include precautions to assure that the sampling apparatus, the testing apparatus and the samples themselves are neither contaminated with, nor altered by any reagent which affects the representativeness of the sample and interferes with the purpose of the work.

14.5 Sample containers shall be of such types as will protect the sample from deterioration and contamination. Specific precautions shall be taken as follows:

a) Sample tubes and other containers shall be tested for leaks and repaired or renewed where necessary. Immediately after filling, the container shall be closed tightly, and when samples are to be kept for 24 hours or more, the closures shall be sealed by dipping in wax or otherwise sealed.

b) Containers shall be cleaned or known to be free from dirt, washing compounds, acids, solvents, soldering fluid, oil or other contaminants. Small parts such as glass rods, rubber tubing connectors and stoppers shall likewise be cleaned and be in good condition.

14.6 Before collecting gas sample the gas shall be allowed to run long enough to purge the lines and also the sample container to obtain a representative and uncontaminated sample for testing. For density determination displacement of saline water can be used to collect the sample without need for purging.

14.7 Sample containers may be made of glass or metal. These are available in a number of different varieties and sizes. Normally a plain glass sample tube has a capacity of 250 or 500 ml, and has a tubulature at each end to connect with rubber tubing, to be closed with pinchcocks, clamps or glass plugs. Metal sample tubes may be made with pipe and fittings; or end fitted with gas cocks which may be welded on. Two commonly used metal tubes are as follow:

a) *Copper Sampling Tube* — 750 ml, 67 mm in diameter, 380 mm overall length with stop cocks at top and bottom ends designed for rubber tubing connectors.

b) *Zinc Sampling Tube* — 2 000 ml, 114 mm diameter, 394 mm overall length with stop cocks at top and bottom ends designed for copper tubing.

14.7.1 Copper and zinc sample containers and sampling tubes are not recommended for petroleum gas, natural gas or other similar gases which contain H_2S or mercaptans as impurity. For these types of gases, use of stainless steel sample containers and sample tubes is recommended.

14.8 To ensure safety the sample container together with all its fittings shall be tested at a pressure one and half times more than the maximum fluid pressure in the pipe line. During this pressure test, the container shall not rupture, deform or crack and shall prove to be totally leak-proof. It is recommended to conduct this pressure test at a temperature equal to the normal fluid temperature of the pipe line.

15. CALIBRATION

15.1 Even though all recommended procedures for setting up the differential-pressure flow meter installations are carefully adhered to in any given case, it is constantly necessary to guard against faulty operation of the system. There shall be a routine schedule of tests and checks which experience has shown are essential for revealing, preventing, and eliminating commonly encountered errors. Although these errors are not inherent in either the primary or secondary element they develop with time under the influence of service conditions and can only be recognised and remedied through continuous inspection. Instructions for adjustment and repair of instruments are furnished by equipment manufacturers, so that difficulties which develop may be readily corrected by the user. However, proper maintenance schedules shall be set up with provisions for frequent zero adjustment and periodic calibration of the flow meter itself, so that operating troubles may be uncovered. The frequency of checking will depend on such considerations as the importance of the measurement, past experience with the type of instrument being used, and how often there are indications of error. In any case, it is recommended to check and adjust the meter for zero setting at least once every day for continuous flow and/or during each interruption for interrupted flow. The meter shall be calibrated at least once in a week.

15.2 A detailed inspection of the condition of the primary element, connecting lines, and attendant facilities shall be made at least once in every 18 months or more frequently, depending on the nature of the gas flowing, and on the agreement between the buyer and the seller. This inspection shall cover a thorough check on all the dimensions of the primary element and shall record alterations if necessary. This is required in addition to the adjustment, cleaning and servicing of the instrument. Attention is drawn to the fact that any alteration in the dimensions of the primary element due to solid deposits or erosion may seriously affect the calibration of the meter.

15.3 Zero Adjustment of the Flow Pen — Checking an instrument for zero setting of the flow pen is a simple procedure which shall be carried out as often as once a day in normal operation. Since changes in temperature affect the zero adjustment, it is desirable to protect the instrument from sunlight or other sources of heat. This effect can be minimized further if the zero adjustment is always made at a time of day when the temperatures are about the same.

In testing the zero setting, the meter shall first be blocked off from the primary element by closing either one (usually the high pressure side) or both of the valves on the take-off connections at the line or primary element. The pressure on the two sides of the instrument is then equalized by opening valves in the by-pass between the connecting lines, making sure that the vent valve in the by-pass remains closed. In high pressure gas meters, it may be desirable to vent both lead lines to the atmosphere while the zero setting is being adjusted. This prevents gas which might be leaking through the block valves at the flange or line taps from flowing through the by pass line with a consequent differential pressure effect on the manometer. When sufficient time has elapsed to permit the manometer fluid to balance, the flow pen should rest at zero on the chart scale. If this is not the case, the pen may be set to zero by means of an adjusting screw on the pen arm or linkage. The meter is then put back into service by reversing the procedure outlined above.

15.4 Calibration of the Flow Pen — A calibration provided a positive means of checking the relation between meter reading and true differential pressure. It is recommended that, whenever possible, meters shall be calibrated where they are to be used rather than in an instrument shop. The flow pen is calibrated by applying to the instrument an accurately known differential pressure and observing if the pen reads correctly on the chart. This is normally done with the help of water columns.

15.4.1 Before starting a calibration, the meter is taken out of service by the procedure described for zero adjustment of the flow pen. After the flow pen is properly zeroed under atmospheric conditions, the water column is connected to the high-pressure side of the meter with the low pressure side open to atmosphere. Air or gas pressure is then applied to the high-pressure side and the water manometer in parallel with it in a test to ensure that there are no leaks in the line joining the pressure legs of the two manometers. The leakage test involves holding a pressure equal to full-scale deflection of the meter for several minutes to see if the meter reading shows any tendency to drop.

15.4.2 After it has been established that there are no leaks in the system, air or gas pressure is again applied to the high pressure side of the meter and the water column in parallel with it. Meter readings are then checked against heights of the water column. In making this calibration test, each point shall be checked twice — once with an increasing differential and once with a decreasing differential. It is important in the former case that the water column reading be approached from below and, in the latter case, from above. In reading the water column, care shall be taken to avoid parallax errors, particularly at high differential pressures. Tapping of the chart, scale, or instrument, or other artificial means of reducing the error during the test are not permissible.

An acceptable calibration is one in which flowmeter readings check the water column within ± 0·5 percent of the total scale or chart range.

15.5 Zero Adjustment of the Pressure Pen — For meter in gas service, an additional pen continuously records static pressure on the same chart with the differential pressure record. The zero setting of this pen shall be checked as frequently as the zero setting of the flow pen. In this case, however, it is important to note that the term zero setting does not necessarily mean 'zero' on the chart, since the zero position depends on the following conditions:

a) Whether the pen is set to record absolute or gauge pressures, and
b) Whether the lead lines are connected to the meter.

If the lead lines are connected to the meter, the effect of fluid head between the primary element and the meter body does not have to be considered. The zero setting for absolute pressure records will be at a chart reading corresponding to the barometric pressure. For gauge pressure records, the zero setting will be at zero on the chart.

The meter is taken out of service with both block values at the line or at the flanges of the primary element closed. The pressures on the system is then slowly released by means of the vent valve in the meter manifold. If necessary, the pressure pen is reset to the correct zero reading by means of a screw adjustment on the pen arm.

15.6 Calibration of the Pressure Pen — It is desirable to check the calibration of the pressure pen at the same time that the flow pen is calibrated. For this purpose, an outlet for a test gauge installation shall be provided in the lead line to which the pressure pen is connected.

15.6.1 The static pressure element of the meter may be tested by means of a water or mercury manometer, a calibrated pressure gauge or a deadweight tester. It is customary to employ manometers for pressure springs whose range is less than 1 atmosphere, calibrated pressure gauges for higher range pressure springs in field testing and dead weight tester in the shop.

15.6.2 In calibrating static pressure element, it should be appreciated that full scale movement of the pen on the chart corresponds to the full pressure range of the spring installed in the meter. Thus charts which are graduated in units of pressure shall have the same range as the spring. If possible, the pressure element shall be checked throughout its range and particularly in the region of normal use. In field tests, however, this check is usually restricted to pressure between atmospheric and the existing pressure of the line in which the primary element is installed.

15.6.3 After the test gauge is in position and the pressure pen is set to its zero reading, the pressure on the meter can be increased slowly in steps by periodic opening and closing of the block valve at the flange of the primary element in the lead line to which the pressure pen is connected. Test gauge readings are then compared with chart readings at several points on the scale between the zero setting and the line pressure. The check is conducted by both increasing and decreasing the pressure and by keeping a record on the chart.

15.6.4 A satisfactory calibration is one in which the pressure pen readings are accurate to within ±0·5 percent of the total scale travel. If results are not acceptable, the static pressure element shall be adjusted (or replaced, if necessary) until the pressure pen reads correctly throughout the range available in the field test.

15.7 Calibration of Thermometers

15.7.1 Calibrating a bimetal thermometer consists of immersing it in baths of known temperatures to assure that for a given temperature change, the pointer has the proper angular deflection. It is sufficient to check this only between two temperatures to get linearity over the entire scale. If the setting of a bimetal thermometer changes, reset the pointer at only one point on the scale. The thermometer will then be accurate over the entire scale.

15.7.2 To calibrate a gas filled thermometer, the bulb shall be placed in a bath at a temperature approximately one-third full scale, and the pen arm set with the zero adjustment to read correctly at this point. The bulb is then placed in a bath having a temperature approximately two-thirds full scale. If the recorder reads within 1 percent of the true temperature, the pen shall be moved to read correctly with the full scale adjustment. The bulb shall then be placed again in the low temperature bath and set with the zero adjustment. Failure of the recorder to read within 1 percent of the true temperature indicates the necessity for a more detailed calibration and setting of the instrument, which may include addition or removal of gas from the bulb.

15.7.3 Liquid filled and vapour tension thermometers can also be calibrated by immersing the bulb in two different temperature baths, as is done for the gas filled thermometers. For bath temperatures, the recorder shall read within the accuracy specified by the manufacturer. This shall be achieved, if necessary, by zero adjustment of the pointer. If the error exceeds the tolerance range and cannot be set by zero adjustment, the thermometer shall be sent for a more detailed check and readjustment.

16. EFFECT OF PRESSURE AND TEMPERATURE ON INSTRUMENT OPERATION

16.1 Since the functioning of flow meters is affected to a very small degree by changes in pressure and temperature, the calibration and zero adjustment of an instrument may be slightly altered under certain conditions. The following variations in chart records can be expected:

a) A variation of the flow pen reading to the extent of about 0·5 cm of water per 35 kgf/cm^2 of pressure,

b) A change in the flow pen zero setting of about 0·33 percent of the total scale range per 10°C variation in temperature, and

c) A change of about 1 percent in pressure pen reading per 10°C variation in temperature.

TABLE 1 MINIMUM STRAIGHT LENGTHS REQUIRED BETWEEN VARIOUS FITTINGS LOCATED UPSTREAM OR DOWNSTREAM OF THE PRIMARY ELEMENT AND THE PRIMARY ELEMENT ITSELF

(*Clauses* 5.3.9, 5.3.11, 5.3.13 *and* 5.4)

β	On Upstream (Inlet) Side of the Primary Element						On Down-stream (Outlet) Side
	Single 90° Bend or Tee (Flow from One Branch Only)	Two or More 90° Bends in the Same Plane	Two or More 90° Bends in Different Planes	Reducer (2 *D* to *D* Over a Length of 3 *D*). Expander (0·5 *D* to *D* Over a Length of 1·5 *D*)	Globe Valve Fully Open	Gate Valve Fully Open	All Fittings Included in this Table
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
≤0·20	10	14	34	16	18	12	4
0·25	10	14	34	16	18	12	4
0·30	10	16	34	16	18	12	5
0·35	12	16	36	16	18	12	5
0·40	14	18	36	16	20	12	6
0·45	14	18	38	18	20	12	6
0·50	14	20	40	20	22	12	6
0·55	16	22	44	20	24	14	6
0·60	18	26	48	22	26	14	7
0·65	22	32	54	24	28	16	7
0·70	28	36	62	26	32	20	7
0·75	36	42	70	28	36	24	8
0·80	46	50	80	30	44	30	8

Fittings	Minimum Upstream (Inlet) Straight Length Required
Abrupt symmetrical reduction having a diameter ratio ≥0·5	30
Thermometer pocket of diameter ≤0·03 *D*	5
Thermometer pocket of diameter between 0·03 *D* and 0·13 *D*	20

Note — This table is valid for all primary elements other than venturi tubes, as defined in this standard. For venturi tubes refer to Table 2. The values given in this table are zero additional tolerance values. All straight lengths are expressed as multiples of the diameter *D*. They should be measured from the upstream face of the primary element.

TABLE 2 MINIMUM STRAIGHT LENGTHS REQUIRED BETWEEN VARIOUS FITTINGS LOCATED UPSTREAM OF THE CLASSICAL VENTURI TUBE AND THE CLASSICAL VENTURI TUBE ITSELF

(*Clauses* 5.3.9, 5.3.11, 5.3.13, 5.5.3, 5.5.4 *and* 11.5.1)

DIAMETER RATIO β	SINGLE 90° SHORT RADIUS BENDS*	TWO OR MORE 90° BENDS IN THE SAME PLANE*	TWO OR MORE 90° BENDS IN DIFFERENT PLANES†	TWO OR MORE 90° BENDS IN DIFFERENT PLANES WITH STRAIGHTENING VANES*	REDUCER 3 D to D OVER A LENGTH OF 3·5 D	EXPANDER 0·75 D to D OVER A LENGTH OF D	GATE VALVE FULLY OPEN
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
0·40	0·5	1·5	—	3·5	2·5	1·5	2·5
0·45	1·0	1·5	—	3·5	4·5	2·5	3·5
0·50	1·5	2·5	—	4·5	5·5	2·5	3·5
0·55	2·5	2·5	—	5·5	6·5	3·5	4·5
0·60	3·0	3·5	—	5·5	8·5	3·5	4·5
0·65	4·0	4·5	—	5·5	9·5	4·5	4·5
0·70	4·0	4·5	—	5·5	10·5	5·5	5·5
0·75	4·5	4·5	—	6·5	11·5	6·5	5·5

NOTE 1 — This table is valid only for classical venturi tubes as defined in this standard having diameter ratios lying between 0·4 and 0·75. All straight lengths are expressed as multiples of the diameter D. Interpolation is permissible.

NOTE 2 — *Straightening Vanes* — The purpose of the vanes is to eliminate swirl and cross-currents set up by the pipe fittings preceding the primary element. Although the following specification apply particularly to the type of vanes constituted by a set of similar circular arc vanes of diameter a and length L arranged symmetrically in the containing pipe and between them allowing for passages of cross-sectional area A, vanes of other designs (square, hexagonal, etc) can be used provided they meet these specifications.

It is not necessary that all the vane passages be of the same size, but their arrangement should be strictly symmetrical.

For the type of vane described above:

a) the maximum transverse dimensions a should not exceed 0·25 D;
b) the cross-sectional area of any passage should not exceed one-sixteenth (1/16) of the cross-sectional area of the containing pipe;
c) the length L of the vanes should be at least 10 a.

The walls of the vane component parts shall be as thin as possible, if necessary reamed at both upstream and downstream ends, and these component parts shall be securely fastened in place so as to prevent them being pushed down against the primary element.

*The radius of curvature of the bend should be equal to or greater than the pipe diameter.

†As the effect of these fittings may still be present after 40 D, no values can be given in the table.

TABLE 3 (LIMITATIONS OF Re_D, D, β, $\frac{P_2}{P_1}$, $\frac{D}{k}$ AND k FOR DIFFERENT TYPES OF PRIMARY ELEMENT)

(*Clauses* 5.3.10 and 5.3.15)

FLOW MEASURING DEVICE	REYNOLDS NUMBER (Re_D)	PIPE DIAMETER (D)	DIAMETER RATIO (β)	PRESSURE RATIO $\frac{P_2}{P_1}$	PIPE ROUGHNESS (k)	$\frac{D}{k}$ RATIO
		mm				
1. *Orifice Plates with:*						
a) Corner taps	*See* Table 7	50-1 000	0·22-0·8 (0·22-0·75)	$\geqslant$ 0·75	—	$\geqslant$ 400
b) Vena contracta taps	*See* Table 10	50-750	0·1-0·8 (0·2-0·7)	$\geqslant$ 0·75	—	$\geqslant$1 000
c) Flange taps	*See* Table 11	50-750	0·1-0·75 (0·2-0·7)	$\geqslant$ 0·75	—	$\geqslant$1 000
2. *Nozzles:*						
a) ISA 1932 nozzles	*See* Table 12	50-1 000	0·45-0·8	$\geqslant$ 0·75	—	$\geqslant$ 400
b) Long radius nozzles	*See* Table 14	50-200	0·2-0·8	$\geqslant$ 0·75	—	$\geqslant$1 000
c) Venturi nozzles	*See* Table 18 or Fig. 15	65-500	0·35-0·75	$\geqslant$ 0·75	—	$\geqslant$ 400
3. *Venturi Tubes with:*						
a) Rough cast convergent	2×10^5 to 2×10^6	100-800	0·3-0·75	$\geqslant$ 0·75	< 0·2	—
b) Machined convergent	2×10^5 to 1×10^6	50-250	0·4-0·75	$\geqslant$ 0·75	< 0·2	—
c) Rough welded sheet iron convergent	2×10^5 to 2×10^6	200-1 200	0·4-0·7	$\geqslant$ 0·75	< 0·2	—

NOTE — The unbracketed terms in the above table are the limiting values. This standard recommends the use of values within brackets for all practical purposes. Refer to **5.3.10**.

TABLE 4 SYMBOLS

(*Clause* 6.1)

SYMBOL	REPRESENTED QUANTITY	DIMENSIONS*
(1)	(2)	(3)
C	Coefficient of discharge, $C = \frac{\alpha}{E}$	Pure number
d	Diameter of throat of the primary element at operating conditions	L
D	Upstream diameter of classical venturi tube (or upstream pipe diameter for orifice plate and nozzles) at operating conditions	L
E	Velocity of approach factor, $E=(1-\beta^4)^{-\frac{1}{2}}$	Pure number
k	Absolute roughness	L
m	Area ratio, $m=\beta^2$	Pure number
P	Absolute static pressure of the fluid	$ML^{-1}T^{-2}$
ΔP	Differential pressure	$ML^{-1}T^{-2}$
q_m	Mass rate of flow	MT^{-1}
q_v	Volume rate of flow	L^3T^{-1}
R	Radius	L
r_o, r_{Re}	Roughness correction factors	Pure number
Re_D	Reynolds number of the flow in the upstream pipe, referred to D	Pure number
t	Temperature of the fluid	Θ
$\tilde{v}$	Mean axial velocity of the fluid in the pipe	LT^{-1}
x	Differential pressure ratio, $x = \frac{\Delta P}{P_1}$	Pure number
X	Acoustic ratio, $X = \frac{x}{\kappa}$	Pure number
α	Flow coefficient	Pure number
β	Diameter ratio, $\beta = \frac{d}{D}$	Pure number
a	Diameter of pressure tap	L
ϵ	Expansibility (expansion) factor	Pure number
η	Dynamic viscosity of the fluid	$ML^{-1}T^{-1}$
κ	Isentropic exponent†	Pure number
υ	Kinematic viscosity of the fluid	L^2T^{-1}
ρ	Mass density of the fluid	ML^3
τ	Pressure ratio, equal to $\frac{P_2}{P_1} = 1 - x$	Pure number
φ	Included angle of the divergent	Pure number
$\Delta\tilde{\omega}$	Pressure loss	$ML^{-1}T^{-2}$

*M=mass L=length T=time Θ= Temperature

†For ideal gases, the ratio of the specific heat capacities and the isentropic exponent have the same values.

Subscript 1 applies to conditions of the fluid in the plane of the upstream pressure tap.

Subscript 2 applies to conditions of the fluid in the plane of the downstream/throat pressure tap.

TABLE 5 COEFFICIENT OF THERMAL EXPANSION θ_m FOR ORIFICE DIAMETER CORRECTION

(*Clause* 8.1.4)

SUBSTANCE (1)	TEMPERATURE (2) °C	$\theta_m \times 10^{-6}$ (3) mm²/mm°C
Aluminium	20-100	23·80
(Commercial)	20-300	26·70
Aluminium bronze	20	17·00
Brass (Commercial)	0-100	19·00
Bronze (Commercial)	25-100	17·50
(Phosphor)	0- 85	16·80
Cast iron	40	10·61
	—190 - +16	8·50
Monel	25-100	13·7-14·50
	25-300	14·9-15·20
Nickel steel		
10 percent Ni	20	13·00
20 percent Ni	20	19·50
30 percent Ni	20	12·00
36 percent Ni (Invar)	20	0·90
40 percent Ni	20	6·00
50 percent Ni	20	9·70
80 percent Ni	20	12·50
Steel (Annealed)	40	10·95
(1·2 percent C)	0-100	10·50
	100-200	11·50
Stainless Steel	20-100	17·28
(Hardened)	20-200	17·64
(Annealed)	20-100	18·54
	20-200	19·26

TABLE 6 SPACING OF VENA CONTRACTA PRESSURE TAPS

(*Clauses* 9.2.4.4 and 9.5.1.1)

β	Spacing of Vena Contracta Pressure Taps
0·10	0·84 D±30%
0·15	0·82 D±30%
0·20	0·80 D±30%
0·25	0·78 D±30%
0·30	0·76 D±30%
0·35	0·73 D±25%
0·40	0·70 D±25%
0·45	0·67 D±25%
0·50	0·63 D±20%
0·55	0·59 D±20%
0·60	0·55 D±15%
0·65	0·50 D±15%
0·70	0·45 D±10%
0·75	0·40 D±10%
0·80	0·34 D±10%

NOTE 1 — The above table gives the spacing of the centre-line of the downstream pressure tap as a function of β, together with the tolerance limits of this spacing.

NOTE 2 — The spacing of the upstream pressure tap from the upstream face of the orifice plate is nominally equal to D and may, without modification in the flow coefficient, be between 0·9 and 1·1 D.

NOTE 3 — The centre-line of any new pressure tap shall always be at the standard spacing, but an existing pressure tap may be used, provided its spacing is between the specified lower and upper limits.

TABLE 7 FLOW COEFFICIENT α_0 FOR ORIFICE PLATES WITH CORNER TAPS

(*Clause* 9.4.1.1)

Re_D / β^4	5×10^3	10^4	2×10^4	3×10^4	5×10^4	10^5	10^6	10^7	β^4
	α_0								
0·0025	0·603	0·600	0·599	0·599	0·598	0·598	0·598	0·597	0·0025
0·003	0·604	0·600	0·600	0·600	0·599	0·599	0·599	0·598	0·003
0·004	0·605	0·601	0·601	0·601	0·600	0·600	0·600	0·599	0·004
0·005	0·606	0·602	0·602	0·602	0·601	0·601	0·600	0·599	0·005
0·01	0·611	0·606	0·605	0·604	0·603	0·603	0·602	0·602	0·01
0·02	0·619	0·613	0·611	0·608	0·607	0·607	0·606	0·606	0·02
0·03	0·627	0·620	0·616	0·613	0·612	0·612	0·611	0·610	0·03
0·04	0·634	0·626	0·621	0·618	0·617	0·616	0·615	0·614	0·04
0·05		0·632	0·626	0·623	0·622	0.620	0·619	0·618	0·05
0·06		0·637	0·631	0·627	0·626	0·624	0·622	0·621	0·06
0·07		0·643	0·636	0·632	0·630	0·628	0·626	0·625	0·07
0·08		0·648	0·641	0·636	0·634	0·632	0·630	0·629	0·08
0·09		0·653	0·646	0·641	0·638	0·636	0·634	0·633	0·09
0·10		0·658	0·650	0·645	0·642	0·640	0·637	0·636	0·10
0·11		0·663	0·655	0·650	0·647	0·644	0·641	0·640	0·11
0·12		0·668	0·659	0·654	0·651	0·647	0·645	0·644	0·12
0·13		0·674	0·664	0·659	0·655	0·651	0·649	0·648	0·13
0·14		0·679	0·668	0·663	0·659	0·655	0·652	0·651	0·14
0·15		0·684	0·673	0·668	0·663	0·659	0·656	0·655	0·15
0·16		0·689	0·677	0·672	0·667	0·663	0·660	0·659	0·16
0·17		0·695	0·682	0·677	0·671	0·667	0·664	0·663	0·17
0·18		0·700	0·687	0·681	0·675	0·671	0·667	0·666	0·18
0·19		0·705	0·692	0·685	0·679	0·675	0·671	0·670	0.19
0·20		0·710	0·696	0·689	0·683	0·679	0·675	0·674	0·20

0·21		0·716	0·701	0·694	0·688	0·683	0·679	0·678	0·21
0·22		0·721	0·705	0·698	0·692	0·687	0·683	0·682	0·22
0·23		0·726	0·710	0·703	0·696	0·691	0·687	0·685	0·23
0·24		0·731	0·714	0·707	0·700	0·695	0·691	0·689	0·24
0·25		0·737	0·719	0·712	0·705	0·699	0·695	0·693	0·25
0·26		0·742	0·723	0·716	0·709	0·703	0·699	0·697	0·26
0·27		0·748	0·728	0·721	0·714	0·708	0·703	0·701	0·27
0·28		0·753	0·733	0·726	0·718	0·712	0·707	0·705	0·28
0·29		0·758	0·738	0·731	0·723	0·716	0·711	0·709	0·29
0·30		0·763	0·743	0·735	0·727	0·720	0·715	0·713	0·30
0·31		0·769	0·748	0·740	0·732	0·725	0·719	0·717	0·31
0·32		0·775	0·753	0·745	0·736	0·729	0·723	0·721	0·32
0·33		0·781	0·759	0·750	0·741	0·734	0·728	0·725	0·33
0·34		0·786	0·764	0·755	0·745	0·738	0·732	0·729	0·34
0·35		0·792	0·770	0·760	0·750	0·743	0·736	0·733	0·35
0·36		0·798	0·775	0·765	0·755	0·748	0·740	0·738	0·36
0·37			0·781	0·770	0·761	0·753	0·744	0·742	0·37
0·38			0·786	0·775	0·766	0·757	0·748	0·747	0·38
0·39			0·792	0·780	0·772	0·762	0·753	0·751	0·39
0·40			0·797	0·786	0·777	0·767	0·757	0·756	0·40
0·41			0·804	0·793	0·783	0·773	0·763	0·761	0·41

NOTE — This type of orifice plate should not be used when

$\beta^4 < 0{\cdot}002\,3$ or $\beta^4 > 0{\cdot}41$ ($\beta < 0{\cdot}22$ or $\beta > 0{\cdot}8$)

$D <$ (50 mm) or $D >$ (1 000 mm)

Re_D be outside of tabulated values.

TABLE 8 VALUES OF r_0 FOR ORIFICE PLATES WITH CORNER TAPS

(*Clause* 9.4.1.2)

D/k β^2	400	800	1 200	1 600	2 000	2 400	2 800	3 200	⩾3 400	β^2
	r_0									
0·1	1·002	1·000	1·000	—	—	—	—	—	1·000	0·1
0·2	1·003	1·002	1·001	1·000	—	—	—	—	1·000	0·2
0·3	1·006	1·004	1·002	1·001	1·000	—	—	—	1·000	0·3
0·4	1·009	1·006	1·004	1·002	1·001	1·000	—	—	1·000	0·4
0·5	1·014	1·009	1·006	1·004	1·002	1·001	1·000	—	1·000	0·5
0·6	1·020	1·013	1·009	1·006	1·003	1·002	1·000	1·000	1·000	0·6
0·64	1·024	1·016	1·011	1·007	1·004	1·002	1·002	1·000	1·000	0·64

NOTE 1 — The value k is a measure of inside pipe wall roughness and is expressed in units of length.

NOTE 2 — Approximate values of k for different materials can be obtained from the various tables given in reference literature. Table 17 gives values of k for a variety of materials, as derived from the Colebrook formula.

NOTE 3 — When the pipe friction coefficient λ has been experimentally determined after measuring the pressure loss with liquid flow, it is possible to derive D/k from the Colebrook formula using appropriate values of λ and Re_D.

TABLE 9 EXPANSIBILITY (EXPANSION) FACTOR FOR ORIFICE PLATES WITH CORNER TAPS

(*Clause* 9.4.1.3)

$\left(\frac{p_2}{p_1}\right)^{\frac{1}{\kappa}}$ \ β^4	0·0	0·05	0·1	0·15	0·20	0·25	0·30	0·35	0·40
Expansibility Factor									
1	1·000 0	1·000 0	1·000 0	1·000 0	1·000 0	1·000 0	1·000 0	1·000 0	1·000 0
0·995	0·997 4	0·997 3	0·997 2	0·997 0	0·996 9	0.996 7	0·996 7	0·996 6	0·996 5
0·99	0·995 0	0·994 8	0·994 6	0·994 3	0·994 1	0·993 7	0·993 7	0·993 5	0·993 3
0·95	0·977 5	0·976 5	0·975 5	0·974 6	0·973 6	0·972 6	0·971 7	0·970 7	0·969 7
0·90	0·956 9	0·955 1	0·953 2	0·951 4	0·949 5	0·947 7	0·945 8	0·944 0	0·942 1
0·80	0·917 7	0·914 1	0·910 6	0·907 1	0·903 5	0·900 0	0·896 5	0·892 9	0·889 4
0·70	0·879 7	0·874 6	0·869 4	0·864 2	0·859 1	0·853 9	0·848 7	0·843 6	0·838 4

TABLE 10 LIMITING VALUES OF REYNOLDS NUMBERS FOR ORIFICE PLATES WITH 'VENA CONTRACTA' TAPS

(*Clause* 9.5.1.3)

D	(50 mm)		(75 mm)		(100 mm)		(150 mm)		(200 mm)		(250 mm)		(375 mm)		(750 mm)	
β^4 | Re_D	Re_D		Re_D		Re_D		Re_D		Re_D		Re_D		Re_D		Re_D	
	Min.	*Max.*	*Min.*	*Max.*	*Min.*	*Max.*	*Min.*	*Max.*	*Min.*	*Max.*	*Min.*	*Max.*	*Min.*	*Max.*	*Min.*	*Max.*
0·100	6 000	10^6	9 000	10^6	12 000	10^6	18 000	10^7	24 000	10^7	30 000	10^7	45 000	10^7	90 000	10^7
0·150	6 000	10^6	9 000	10^6	12 000	10^6	18 000	10^7	24 000	10^7	30 000	10^7	45 000	10^7	90 000	10^7
0·200	6 000	10^6	9 000	10^6	12 000	10^6	18 000	10^7	24 000	10^7	30 000	10^7	45 000	10^7	90 000	10^7
0·250	6 000	10^6	9 000	10^6	12 000	10^6	18 000	10^7	24 000	10^7	30 000	10^7	45 000	10^7	90 000	10^7
0·300	6 000	10^6	9 000	10^6	12 000	10^6	18 000	10^7	24 000	10^7	30 000	10^7	45 000	10^7	90 000	10^7
0·325	6 000	10^6	9 000	10^6	12 000	10^6	18 000	10^7	24 000	10^7	30 000	10^7	45 000	10^7	90 000	10^7
0·350	6 000	10^6	9 000	10^6	12 000	10^6	20 000	10^7	25 000	10^7	30 000	10^7	50 000	10^7	100 000	10^7
0·375	6 000	10^6	10 000	10^6	15 000	10^6	25 000	10^7	25 000	10^7	30 000	10^7	50 000	10^7	100 000	10^7
0·400	8 000	10^6	10 000	10^6	15 000	10^6	25 000	10^7	30 000	10^7	50 000	10^7	50 000	10^7	100 000	10^7
0·425	8 000	10^6	15 000	10^6	20 000	10^6	30 000	10^7	30 000	10^7	50 000	10^7	50 000	10^7	150 000	10^7
0·450	10 000	10^6	15 000	10^6	20 000	10^6	30 000	10^7	50 000	10^7	50 000	10^7	50 000	10^7	150 000	10^7
0·475	10 000	10^6	15 000	10^6	20 000	10^6	30 000	10^7	50 000	10^7	50 000	10^7	100 000	10^7	200 000	10^7
0·500	10 000	10^6	20 000	10^6	20 000	10^6	50 000	10^7	50 000	10^7	50 000	10^7	100 000	10^7	200 000	10^7
0·520	15 000	10^6	20 000	10^6	30 000	10^6	50 000	10^7	50 000	10^7	50 000	10^7	100 000	10^7	200 000	10^7
0·540	15 000	10^6	20 000	10^6	30 000	10^6	50 000	10^7	50 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7
0·560	15 000	10^6	25 000	10^6	30 000	10^6	50 000	10^7	50 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7
0·580	15 000	10^6	25 000	10^6	30 000	10^6	50 000	10^7	100 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7
0·600	20 000	10^6	25 000	10^6	30 000	10^6	50 000	10^7	100 000	10^7	100 000	10^7	100 000	10^7	300 000	10^7
0·620	20 000	10^6	25 000	10^6	50 000	10^6	50 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7	300 000	10^7
0·640	20 000	10^6	25 000	10^6	50 000	10^6	50 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7	300 000	10^7
0·660	20 000	10^6	25 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7	300 000	10^7
0·680	20 000	10^6	25 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7	300 000	10^7
0·700	25 000	10^6	50 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7	400 000	10^7
0·720	25 000	10^6	50 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	200 000	10^7	200 000	10^7	400 000	10^7
0·740	25 000	10^6	50 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	200 000	10^7	200 000	10^7	400 000	10^7
0·750	25 000	10^6	50 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	200 000	10^7	200 000	10^7	400 000	10^7
0·760	25 000	10^6	50 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	200 000	10^7	200 000	10^7	400 000	10^7
0·770	25 000	10^6	50 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	200 000	10^7	200 000	10^7	400 000	10^7
0·780	25 000	10^6	50 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	200 000	10^7	200 000	10^7	400 000	10^7
0·790	25 000	10^6	50 000	10^6	50 000	10^6	100 000	10^7	200 000	10^7	200 000	10^7	200 000	10^7	400 000	10^7
0·800	50 000	10^6	50 000	10^6	100 000	10^6	100 000	10^7	200 000	10^7	200 000	10^7	200 000	10^7	500 000	10^7

TABLE 11 LIMITING VALUES OF REYNOLDS NUMBERS FOR ORIFICE PLATES WITH FLANGE TAPS

(*Clause* 9.6.1.2)

D	50 mm		75 mm		100 mm		150 mm		200 mm		250 mm		375 mm		750 mm	
Re_D	Re_D		Re_D		Re_D		Re_D		Re_D		Re_D		Re_D		Re_D	
β	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
0·100	8 000	10^6	12 000	10^6	16 000	10^6	24 000	10^7	32 000	10^7	40 000	10^7	60 000	10^7	120 000	10^7
0·150	8 000	10^6	12 000	10^6	16 000	10^6	24 000	10^7	32 000	10^7	40 000	10^7	60 000	10^7	120 000	10^7
0·200	8 000	10^6	12 000	10^6	16 000	10^6	24 000	10^7	32 000	10^7	40 000	10^7	60 000	10^7	120 000	10^7
0·250	8 000	10^6	12 000	10^6	16 000	10^6	24 000	10^7	32 000	10^7	40 000	10^7	60 000	10^7	120 000	10^7
0·300	8 000	10^6	12 000	10^6	16 000	10^6	24 000	10^7	32 000	10^7	40 000	10^7	60 000	10^7	120 000	10^7
0·350	8 000	10^6	12 000	10^6	16 000	10^6	24 000	10^7	32 000	10^7	40 000	10^7	60 000	10^7	120 000	10^7
0·400	8 000	10^6	12 000	10^6	16 000	10^6	30 000	10^7	40 000	10^7	40 000	10^7	60 000	10^7	120 000	10^7
0·450	8 000	10^6	15 000	10^6	20 000	10^6	30 000	10^7	50 000	10^7	40 000	10^7	75 000	10^7	150 000	10^7
0·500	8 000	10^6	20 000	10^6	30 000	10^6	50 000	10^7	75 000	10^7	75 000	10^7	100 000	10^7	200 000	10^7
0·550	10 000	10^6	20 000	10^6	30 000	10^6	50 000	10^7	75 000	10^7	75 000	10^7	100 000	10^7	200 000	10^7
0·600	20 000	10^6	30 000	10^6	40 000	10^6	50 000	10^7	75 000	10^7	100 000	10^7	200 000	10^7	300 000	10^7
0·625	20 000	10^6	30 000	10^6	40 000	10^6	100 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7	300 000	10^7
0·650	30 000	10^6	30 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7	300 000	10^7
0·675	30 000	10^6	40 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	100 000	10^7	200 000	10^7	300 000	10^7
0·700	50 000	10^6	40 000	10^6	50 000	10^6	100 000	10^7	100 000	10^7	200 000	10^7	200 000	10^7	400 000	10^7
0·725				10^6	50 000	10^6	100 000	10^7	100 000	10^7	200 000	10^7	500 000	10^7	400 000	10^7
0·750				10^6	50 000	10^6	100 000	10^7	500 000	10^7	200 000	10^7	500 000	10^7	400 000	10^7

TABLE 12 FLOW COEFFICIENT α' FOR ISA 1932 NOZZLES

(*Clause* 10.1.7.1)

Re_D / β^4	$2 \cdot 18^4$	$3 \cdot 10^4$	$5 \cdot 10^4$	$7 \cdot 10^4$	10^5	16^6	Re_D / β^4
0·01				0·989	0·989	0·989	0·01
0·02				0·993	0·993	0·993	0·02
0·03				0·996	0·996	0·997	0·03
0·04	0·981	0·989	0·996	0·999	1·000	1·000	0·04
0·05	0·983	0·991	0·998	1·001	1·002	1·003	0·05
0·06	0·985	0·993	1·000	1·004	1·005	1·006	0·06
0·07	0·987	0·995	1·003	1·007	1·008	1·009	0·07
0·08	0·990	0·998	1·006	1·010	1·011	1·012	0·08
0·09	0·993	1·001	1·009	1·013	1·014	1·016	0·09
0·10	0·997	1·004	1·012	1·016	1·017	1·020	0·10
0·11	1·001	1·008	1·016	1·020	1·021	1·024	0·11
0·12	1·005	1·012	1·020	1·024	1·025	1·028	0·12
0·13	1·009	1·016	1·023	1·028	1·029	1·031	0·13
0·14	1·014	1·020	1·027	1·031	1·033	1·035	0·14
0·15	1·018	1·024	1·031	1·035	1·037	1·039	0·15
0·16	1·023	1·028	1·035	1·039	1·041	1·043	0·16
0·17	1·027	1·032	1·039	1·043	1·045	1·047	0·17
0·18	1·032	1·037	1·043	1·047	1·049	1·051	0·18
0·19	1·037	1·041	1·047	1·051	1·053	1·055	0·19
0·20	1·042	1·046	1·052	1·055	1·057	1·059	0·20
0·21	1·047	1·051	1·056	1·059	1·061	1·063	0·21
0·22	1·053	1·057	1·061	1·064	1·065	1·067	0·22
0·23	1·058	1·062	1·066	1·069	1·070	1·071	0·23
0·24	1·064	1·068	1·071	1·073	1·074	1·076	0·24

0·25	1·070	1·074	1·076	1·078	1·079	1·081	0·25
0·26	1·077	1·080	1·081	1·083	1·084	1·085	0·26
0·27	1·083	1·086	1·087	1·088	1·089	1·090	0·27
0·28	1·090	1·092	1·092	1·093	1·094	1·095	0·28
0·29	1·097	1·098	1·098	1·099	1·099	1·100	0·29
0·30	1·104	1·104	1·104	1·104	1·104	1·105	0·30
0·31	1·111	1·111	1·110	1·110	1·110	1·110	0·31
0·32	1·118	1·118	1·116	1·116	1·115	1·115	0·32
0·33	1·125	1·125	1·123	1·122	1·121	1·121	0·33
0·34	1·133	1·132	1·129	1·128	1·127	1·126	0·34
0·35	1·141	1·139	1·136	1·134	1·133	1·132	0·35
0·36	1·149	1·146	1·142	1·140	1·139	1·138	0·36
0·37	1·157	1·154	1·149	1·147	1·146	1·144	0·37
0·38	1·165	1·161	1·156	1·154	1·152	1·150	0·38
0·39	1·173	1·169	1·163	1·161	1·159	1·156	0·39
0·40	1·182	1·177	1·171	1·168	1·166	1·163	0·40
0·41	1·191	1·185	1·179	1·176	1·173	1·170	0·41

NOTE 1 — α'' is not dependent on D.

NOTE 2 — This type of nozzle shall not be used when—

$\beta^4 < 0{\cdot}01$ or $\beta^4 > 0{\cdot}41$ ($\beta < 0{\cdot}32$ or $\beta > 0{\cdot}8$)

$D < 50$ mm or $D > 1\,000$ mm

Re_D is outside the tabulated values.

TABLE 13 VALUES OF r_0 FOR ISA 1932 AND VENTURI NOZZLES

(*Clauses* 10.1.7.2 and 12.6.2)

D/k / β^2	400	800	1 200	1 600	2 000	2 400	2 800	⩾ 3 200
	r_0							
0·3	1·002	1·000	1·000	1·000	1·000	1·000	1·000	1·000
0·4	1·003	1·002	1·000	1·000	1·000	1·000	1·000	1·000
0·5	1·008	1·005	1·003	1·002	1·000	1·000	1·000	1·000
0·6	1·014	1·009	1·006	1·004	1·002	1·001	1·000	1·000
0·65	1·016	1·012	1·009	1·007	1·005	1·003	1·002	1·000

TABLE 14 LONG RADIUS NOZZLES

(*Clause* 10.2.6.1)

Re_D / β	6 000	7 000	8 000	9 000	10 000	12 000	15 000	20 000	25 000	30 000	50 000	10^5	$2 \cdot 10^5$	$5 \cdot 10^5$	10^6	$2 \cdot 10^6$	10^7	Re_D / β
0·20	0·962	0·965	0·967	0·969	0·970	0·972	0·975	0·978	0·980	0·982	0·986	0·990	0·992					0·20
0·25	0·957	0·960	0·963	0·965	0·966	0·969	0·972	0·975	0·978	0·979	0·984	0·988	0·991					0·25
0·30	0·955	0·958	0·960	0·962	0·964	0·967	0·970	0·974	0·976	0·978	0·982	0·987	0·990	0·994				0·30
0·35	0·953	0·956	0·959	0·961	0·962	0·965	0·969	0·972	0·975	0·977	0·981	0·986	0·990	0·993	0·995			0·35
0·40	0·951	0·954	0·957	0·959	0·961	0·964	0·967	0·971	0·974	0·976	0·980	0·986	0·989	0·993	0·995	0·996		0·40
0·45	0·949	0·953	0·955	0·957	0·959	0·962	0·966	0·970	0·972	0·975	0·980	0·985	0·989	0·992	0·994	0·996		0·45
0·50	0·948	0·951	0·954	0·956	0·958	0·961	0·964	0·968	0·971	0·973	0·979	0·984	0·988	0·992	0·994	0·995		0·50
0·55	0·946	0·949	0·952	0·954	0·956	0·959	0·963	0·967	0·970	0·972	0·978	0·983	0·987	0·991	0·994	0·995		0·55
0·60	0·944	0·947	0·950	0·952	0·954	0·958	0·961	0·966	0·969	0·971	0·976	0·982	0·987	0·991	0·993	0·995	0·997	0·60
0·65	0·942	0·945	0·948	0·951	0·953	0·956	0·960	0·964	0·967	0·970	0·975	0·981	0·986	0·990	0·993	0·995	0·997	0·65
0·70	0·940	0·943	0·946	0·949	0·951	0·954	0·958	0·963	0·966	0.968	0·974	0·980	0·985	0·990	0·992	0·994	0·997	0·70
0·75	0·938	0·941	0·944	0·947	0·949	0·953	0·957	0·961	0·965	0·967	0·973	0·979	0·984	0·989	0·992	0·994	0·997	0·75
0·80	0·935	0·939	0·942	0·945	0·947	0·951	0·955	0·960	0·963	0·966	0·972	0·978	0·984	0·989	0·991	0·993	0·996	0·80

TABLE 15 EXPANSIBILITY (EXPANSION) FACTOR ϵ FOR VENTURI TUBES FOR GASES IN TERMS OF PRESSURE RATIO $\frac{P_2}{P_1}$, β^4 AND ISENTROPIC EXPONENT κ*

(*Clause* 11.8)

$\frac{P_2}{P_1}$		1·0	0·98	0·96	0·94	0·92	0·90	0·85	0·80	0·73
β^2	β^4	ϵ for κ=1·2								
0	0	1·0	0·987 4	0·974 8	0·962 0	0·949 1	0·936 1	0·902 9	0·868 9	0·834 0
0·316 2	0·1	1·0	0·985 6	0·971 2	0·956 8	0·942 3	0·927 8	0·891 3	0·854 3	0·816 9
0·447 2	0·2	1·0	0·983 4	0·966 9	0·950 4	0·934 1	0·917 8	0·877 3	0·837 1	0·797 0
0·547 7	0·3	1·0	0·980 5	0·961 3	0·942 4	0·923 8	0·905 3	0·860 2	0·816 3	0·773 3
0·632 5	0·4	1·0	0·976 7	0·954 1	0·932 0	0·910 5	0·889 5	0·839 0	0·790 9	0·744 8
0·640 3	0·41	1·0	0·976 3	0·953 2	0·930 8	0·909 0	0·887 7	0·836 6	0·788 1	0·741 6
ϵ for κ=1·3										
0	0	1·0	0·988 4	0·976 7	0·964 9	0·952 9	0·940 8	0·910 0	0·878 3	0·845 7
0·316 2	0·1	1·0	0·986 7	0·973 4	0·960 0	0·946 6	0·933 1	0·899 0	0·864 5	0·829 4
0·447 2	0·2	1·0	0·984 6	0·969 3	0·954 1	0·938 9	0·923 7	0·885 9	0·848 1	0·810 2
0·547 7	0·3	1·0	0·982 0	0·964 2	0·946 6	0·929 2	0·912 0	0·869 7	0·828 3	0·787 5
0·632 5	0·4	1·0	0·978 5	0·957 5	0·936 9	0·916 8	0·897 1	0·849 5	0·803 9	0·759 9
0·640 3	0·41	1·0	0·978 1	0·956 7	0·935 8	0·915 4	0·895 4	0·847 2	0·801 2	0·756 9

ε for κ=1·4

0	0	1·0	0·989 2	0·978 3	0·967 3	0·956 2	0·944 9	0·916 2	0·886 5	0·855 8
0·316 2	0·1	1·0	0·987 7	0·975 3	0·962 8	0·950 3	0·937 7	0·905 8	0·873 3	0·840 2
0·447 2	0·2	1·0	0·985 7	0·971 5	0·957 3	0·943 0	0·928 8	0·893 3	0·857 7	0·821 9
0·547 7	0·3	1·0	0·983 3	0·966 7	0·950 3	0·934 0	0·917 8	0·878 0	0·838 8	0·800 0
0·632 5	0·4	1·0	0·980 0	0·960 4	0·941 2	0·922 3	0·903 8	0·858 8	0·815 4	0·773 3
0·640 3	0·41	1·0	0·979 6	0·959 6	0·940 1	0·920 9	0·902 1	0·856 6	0·812 7	0·770 4

ε for κ=1·66

0	0	1·0	0·990 9	0·981 7	0·972 4	0·962 9	0·953 3	0·928 8	0·903 3	0·876 8
0·316 2	0·1	1·0	0·989 6	0·979 1	0·968 5	0·957 8	0·947 1	0·919 7	0·891 7	0·862 9
0·447 2	0·2	1·0	0·987 9	0·975 9	0·963 7	0·951 6	0·939 4	0·908 8	0·877 8	0·846 4
0·547 7	0·3	1·0	0·985 8	0·971 8	0·957 7	0·943 8	0·929 9	0·895 3	0·860 9	0·826 5
0·632 5	0·4	1·0	0·983 1	0·966 4	0·949 9	0·933 6	0·917 6	0·878 2	0·839 7	0·802 0
0·640 3	0·41	1·0	0·982 7	0·965 7	0·949 0	0·932 4	0·916 1	0·876 2	0·837 3	0·799 3

*The values for $\beta^2 = \beta^4 = 0$ and $\frac{P_2}{P_1} = 1$ are only given in order to make it possible to interpolate for values of ε for $\beta^4 < 0{\cdot}1$ and $\frac{P_2}{P_1} > 0{\cdot}98$.

The fact this values are given to four figures does not indicate the accuracy of ε.

TABLE 16 FLOW COEFFICIENT α_0 FOR VENTURI NOZZLES

(*Clause* 12.6.1)

β^2	β^4	α_0	β^2	β^4	α_0
0·1	0·01	0·989	0·447 2	0·20	1·066
0·141 4	0·02	0·993	0·458 3	0·21	1·070
0·173 2	0·03	0·997	0·469 0	0·22	1·075
0·200 0	0·04	1·001	0·479 6	0·23	1·080
0·223 6	0·05	1·005	0·489 9	0·24	1·086
0·244 9	0·06	1·009	0·500 0	0·25	1·091
0·264 6	0·07	1·013	0·509 9	0·26	1·096
0·282 8	0·08	1·016	0·519 6	0·27	1·102
0·300 0	0·09	1·020	0·529 2	0·28	1·107
			0·538 5	0·29	1·113
0·316 2	0·10	1·024			
0·331 7	0·11	1·027	0·547 7	0·30	1·119
0·346 4	0·12	1·031	0·556 8	0·31	1·124
0·360 6	0·13	1·035	0·565 7	0·32	1·130
0·374 2	0·14	1·039	0·574 5	0·33	1·136
0·387 3	0·15	1·043	0·583 1	0·34	1·142
0·400 0	0·16	1·048	0·591 6	0·35	1·149
0·412 3	0·17	1·052	0·600 0	0·36	1·155
0·424 3	0·18	1·057			
0·435 9	0·19	1·061			

TABLE 17 VALUES OF THE PIPE WALL ROUGHNESS k

(*Clause* 12.6.2)

MATERIAL (1)	CONDITION (2)	k IN mm (3)
Brass, copper, aluminium, plastics, glass	smooth, without sediments	<0·03
Steel	New, seamless cold drawn	<0·03
	New, seamless hot drawn New, seamless rolled New, welded longitudinally	0·05 to 0·10
	New, welded spirally	0·10
	Slightly rusted	0·10 to 0·20
	Rusty	0·20 to 0·30
	Encrusted	0·50 to 2
	With heavy incrustations	>2
	Bituminized, new	0·03 to 0·05
	Bituminized, normal	0·10 to 0·20
	Galvanized	0·13
Cast iron	New	0·25
	Rusty	1·0 to 1·5
	Encrusted	>1·5
	Bituminized, new	0·1 to 0·15
Asbestos cement	insulated and not insulated, new	<0·03
	not insulated, normal	0·05

TABLE 18 LIMITING VALUES OF REYNOLDS NUMBER Re_D FOR FLOW COEFFICIENTS OF VENTURI NOZZLES

(*Clauses* 12.6.1 *and* 12.6.3)

β^2	β^4	Re_D *Min*	Re_D *Max*
(1)	(2)	(3)	(4)
0.100 0	0.01	0.75×10^5	0.75×10^6
0.141 4	0.02	0.90×10^5	0.85×10^6
0.173 2	0.03	1.00×10^5	0.93×10^6
0.200 0	0.04	1.08×10^5	1.00×10^6
0.223 6	0.05	1.13×10^5	1.06×10^6
0.244 9	0.06	1.18×10^5	1.11×10^6
0.264 6	0.07	1.23×10^5	1.16×10^6
0.282 8	0.08	1.27×10^5	1.21×10^6
0.300 0	0.09	1.31×10^5	1.25×10^6
0.316 2	0.10	1.35×10^5	1.29×10^6
0.331 7	0.11	1.38×10^5	1.33×10^6
0.346 4	0.12	1.42×10^5	1.37×10^6
0.360 6	0.13	1.45×10^5	1.40×10^6
0.374 2	0.14	1.48×10^5	1.43×10^6
0.387 3	0.15	1.51×10^5	1.47×10^6
0.400 0	0.16	1.54×10^5	1.50×10^6
0.412 3	0.17	1.57×10^5	1.53×10^6
0.424 3	0.18	1.60×10^5	1.56×10^6
0.435 9	0.19	1.62×10^5	1.59×10^6
0.447 2	0.20	1.65×10^5	1.62×10^6
0.458 3	0.21	1.67×10^5	1.64×10^6
0.469 0	0.22	1.70×10^5	1.67×10^6
0.479 6	0.23	1.72×10^5	1.70×10^6
0.489 9	0.24	1.75×10^5	1.72×10^6
0.500 0	0.25	1.77×10^5	1.75×10^6
0.509 9	0.26	1.79×10^5	1.77×10^6
0.519 6	0.27	1.81×10^5	1.80×10^6
0.529 2	0.28	1.84×10^5	1.82×10^6
0.538 5	0.29	1.86×10^5	1.84×10^6
0.547 7	0.30	1.88×10^5	1.87×10^6
0.556 8	0.31	1.90×10^5	1.89×10^6
0.565 7	0.32	1.92×10^5	1.91×10^6
0.574 5	0.33	1.94×10^5	1.93×10^6
0.583 1	0.34	1.96×10^5	1.96×10^6
0.591 6	0.35	1.98×10^5	1.98×10^6
0.600 0	0.36	2.00×10^5	2.00×10^6

APPENDIX A

EXAMPLES FOR CALCULATION OF FLOW RATE

A-1. GENERAL DATA AVAILABLE

a) Upstream pipe diameter at operating conditions $= D = 102{\cdot}23$ mm

b) Absolute roughness of the pipe $= k = 0{\cdot}04$ mm

c) Coefficient of thermal expansion for the material of the primary element $= \theta_m = 1{\cdot}1 \times 10^{-5} \dfrac{\text{m}}{\text{m}^\circ\text{C}}$

d) Absolute static pressure of the gas on the upstream side of the primary element $= P_1 = 3{\cdot}845\,6$ kg/cm² absolute $= 377\,160{\cdot}1 \dfrac{\text{kg m}}{\text{sec}^2\ \text{m}^2}$

e) Temperature of the gas flowing through the primary element $= T = 50 + 273 = 323$ K

f) Standard pressure $= P_b = 1{\cdot}033\,2$ kg/cm² absolute $= 101\,322{\cdot}3 \dfrac{\text{kg m}}{\text{s}^2\ \text{m}^2}$

g) Standard temperature $= T_b = 15 + 273 = 288$ K

h) Molecular weight of the gas flowing $= M = 25$

j) Density of the gas at 0°C (or 273 K) and 1·0332 kg/cm²

$$= \left(\frac{\text{Mol mass}}{22{\cdot}4140}\ \text{kg/m}^3\right) = \rho = \frac{25}{22{\cdot}414\,0}\ \frac{\text{kg}}{\text{m}^3}$$

k) Density of the gas at STP

$$= \frac{25}{22{\cdot}4140} \times \frac{273}{288} = \rho_b = 1{\cdot}057\,2 \frac{\text{kg}}{\text{m}^3}$$

m) Density of the gas at flow temperature and pressure condition

$$= 1{\cdot}0572 \times \frac{288}{323} \times \frac{3{\cdot}8456}{1{\cdot}0332} = \rho_1 = 3{\cdot}508\,3 \frac{\text{kg}}{\text{m}^3}$$

n) Absolute viscosity of the gas at upstream flow conditions $= \eta = 1{\cdot}4 \times 10^{-5} \dfrac{\text{kg}}{\text{m sec}}$

p) Kinematic viscosity of the gas at the flow condition $= v = 3{\cdot}990\,5 \times 10^{-6}\,\dfrac{\text{m}^2}{\text{s}}$

q) Isentropic exponent of the gas $= \kappa = 1{\cdot}21$

r) Reynolds number

$$= \frac{\overline{V}_1 D}{v} \text{ (from clause \textbf{6.4.2})}$$

$$= \frac{4q_\text{m}}{\pi\,\rho_1\,Dv_1} = \frac{4\,q_\text{m} \times 10^6}{(3{\cdot}141\,6)(3{\cdot}508\,3)(0{\cdot}102\,23)(3{\cdot}990\,6)} = Re_\text{D}$$

$$= 8{\cdot}8963 \times 10^5\,q_\text{m}$$

A-2. FLOW MEASUREMENT WITH ORIFICE METER (FLANGE TAPS)

A-2.1 Additional Data Available

a) Diameter of the orifice at standard temperature [$\rho(15^\circ\text{C})$] $= d_\text{m} = 48{\cdot}30$ mm

b) Diameter of the orifice at operating conditions $= d_\text{m}[1 + \theta_\text{m}(T - T_\text{b})]$ (*see* clause **8.1.4**)

$= 0{\cdot}048\,26\,[1 + 1{\cdot}1 \times 10^{-5}\,(323\text{-}288)] = d = 48{\cdot}30$ mm

c) Diameter ratio $= \dfrac{d}{D} = \dfrac{48{\cdot}30}{102{\cdot}23} = \beta = 0{\cdot}472\,4$

d) Absolute static pressure of the gas on the downstream side of the orifice $= P_2 = 3{\cdot}721\,2\,\dfrac{\text{kg}}{\text{cm}^2}$

e) Differential pressure $P_1 - P_2$ $= \Delta P = 0{\cdot}124\,4$ kg/cm²

$= 12\,199{\cdot}5\,\dfrac{\text{kg m}}{\text{s}^2\,\text{m}^2}$

f) Differential pressure ratio $= \dfrac{\Delta P}{P_1} = x = 0{\cdot}032\,3$

g) Acoustic ratio $= \dfrac{x}{\kappa} = X = 0{\cdot}026\,7$

h) Pressure ratio $= \dfrac{P_2}{P_1} = 1 - x = \tau = 0{\cdot}967\,6$

A-2.2 Computation of Flow Coefficient, α

From **9.6.1.1** :

$$\alpha = \alpha' \left(1 + \frac{\beta A}{Re_\text{D}}\right)$$

where $\alpha = \alpha_\text{e}\left[\dfrac{10^6 d}{10^6 d + 381A}\right]$

$$
\begin{aligned}
\text{Now } \alpha_e &= 0{\cdot}599\,3 + \frac{0{\cdot}178\,8}{D} + \left[0{\cdot}364 + \frac{0{\cdot}383\,0}{\sqrt{D}}\right]\beta^4 \\
&= 0{\cdot}4\left(1{\cdot}6 - \frac{25{\cdot}40}{D}\right)^5 \left[\left(0{\cdot}07 + \frac{12{\cdot}70}{D}\right) - \beta\right]^{5/2} \\
&\quad - \left(0{\cdot}009 + \frac{0{\cdot}863\,6}{D}\right)\left(0{\cdot}5 - \beta\right)^{3/2} \\
&\quad + \left(\frac{41\,935}{D^2} + 3\right)(\beta - 0{\cdot}7)^{5/2}
\end{aligned}
$$

$\beta = 0{\cdot}472\,4$, therefore $\beta^2 = 0{\cdot}223\,2$, $\beta^3 = 0{\cdot}105\,4$ and $\beta^4 = 0{\cdot}049\,8$

$D = 102{\cdot}235$ mm, therefore $D^2 = 10\,451{\cdot}995$ and $\sqrt{D} = 10{\cdot}111\,1$

$(\beta - 0{\cdot}7)$ is a negative quantity, therefore $(\beta - 0{\cdot}7)^{5/2}$ is an imaginary number and equal to zero (*see* Notes under **9.6.1.1**) or,

$$
\begin{aligned}
\alpha_e = {} & 0{\cdot}599\,3 + 0{\cdot}001\,74 + 0{\cdot}018\,86 + 0{\cdot}4\,(1{\cdot}351\,6)^5\,(0{\cdot}194\,22 - 0{\cdot}472\,4)^{5/2} \\
& - (0{\cdot}017\,45)(0{\cdot}004\,6)
\end{aligned}
$$

The fourth term in the above equation is also an imaginary number and can be considered as Zero

Therefore $\alpha_e = 0{\cdot}62$

$$A = 0{\cdot}039\,37\, d\left(830 - 5\,000\,\beta + 9\,000\,\beta^2 - 4\,200\,\beta + \frac{2\,671}{\sqrt{D}}\right)$$

or $A - 567{\cdot}2$

$$\text{Therefore, } \alpha' = 0{\cdot}62\left(\frac{10^6 \times 48{\cdot}26}{10^6 \times 48{\cdot}26 + 381 \times 567{\cdot}2}\right)$$

$$\text{or } \alpha' = 0{\cdot}617\,2$$

And from **A-1** (r) $Re_D = 8{\cdot}896\,3 \times 10^5 q_m$

$$\text{Therefore, } \alpha = 0{\cdot}617\,2\left[1 + \frac{0{\cdot}472\,4 \times 567{\cdot}2}{8{\cdot}896\,3 \times 10^5 q_m}\right]$$

$$\text{or } \alpha = 0{\cdot}617\,2\left[1 + \frac{3{\cdot}011\,9 \times 10^{-4}}{q_m}\right] \qquad \text{.. (Eqn A)}$$

A-2.3 Computation of ϵ

$\epsilon = 1 - (0{\cdot}41 + 0{\cdot}35\,\beta^4)\,X$

$\beta^4 = 0{\cdot}049\,8$ and $X = 0{\cdot}026\,7$

or $\epsilon = 0{\cdot}988\,6$.

A-2.4 From **8.1.1** :

$$q_m = \alpha\,\epsilon\,\frac{\pi}{4}\,d^2\sqrt{2\,\triangle P\,\rho_1}$$

$$= \alpha\; 0{\cdot}988\,6\,\frac{3{\cdot}141\,6}{4}\,(0{\cdot}048\,3)^2\sqrt{2\,(12\,199{\cdot}5)\,(3{\cdot}508\,3)}$$

$$\text{or } q_m = 0{\cdot}529\,9\,\alpha \qquad \text{.. (Eqn B)}$$

TRIAL 1

Let $\alpha = 0{\cdot}617\,5$

From Eqn (B) $q_m = 0{\cdot}327\,2$ kg/s

Substituting this value of q_m in Eqn (A)

$$= 0{\cdot}617\,2\left[1 + \frac{3{\cdot}011\,9 \times 10^{-4}}{0{\cdot}327\,2}\right]$$

$= 0{\cdot}617\,8 \neq$ Assumed value of α

TRIAL 2

Let $\alpha = 0{\cdot}617\,8$

From Eqn (B) $q_m = 0{\cdot}327\,4$ kg/s

Substituting this value of q_m in Eqn (A)

$$\alpha = 0{\cdot}617\,2\left[1 + \frac{3{\cdot}011\,9 \times 10^{-4}}{0{\cdot}327\,4}\right]$$

$= 0{\cdot}617\,8 =$ Assumed value.

Therefore, the flow for $0{\cdot}124\,4$ kg/cm² pressure differential

$q_m = 0{\cdot}327\,4$ kg/s

and $q_v = \frac{q_m}{\rho_b} = \frac{0{\cdot}327\,4}{1{\cdot}057\,2}\frac{\text{m}^3}{\text{s}}$ at STP

$= 0{\cdot}309\,7\frac{\text{m}^3}{\text{s}}$ at STP

A-2.5 For This Problem

a) $k = 0{\cdot}04$ mm

b) $\frac{P_2}{P_1} = 0{\cdot}967\,6$ and is $> 0{\cdot}75$

c) $\frac{D}{k} = \frac{102{\cdot}23}{0{\cdot}04} = 2\,555{\cdot}75 > 1\,000$

d) $D = 102{\cdot}23$ mm and is between 50 and 760 mm

e) $Re_D = 8{\cdot}896\,3 \times 10^5 \times 0{\cdot}327\,4$
$= 2{\cdot}912\,6 \times 10^5$

and is within the limits specified for orifice plates with flange taps (*see* Table 11). Therefore all the requirements of clauses **9.6.1.1** to **9.6.1.3**, are fulfilled and the procedure as followed in **A-2.2**, **A-2.3** and **A-2.4** is valid.

A-3. FLOW MEASUREMENT WITH NOZZLE ISA 1932 TYPE

A-3.1 Additional Data Available

a) Diameter of the throat at operating condition $= d = 40{\cdot}89$ mm

b) Diameter ratio $\frac{d}{D} = \frac{40{\cdot}89}{102{\cdot}23} = \beta = 0{\cdot}4$

c) Absolute static pressure of the gas on the down stream side of the nozzle $= P_2 = 3{\cdot}746\,6$ kg/cm^2

d) Differential pressure $P_1 - P_2 = \triangle P = 0{\cdot}099$ kg/cm^2 $= 9\,708{\cdot}6 \frac{\text{kg m}}{\text{s}^2\text{m}^2}$

e) Pressure ratio $\frac{P_2}{P_1} = \tau = 0{\cdot}974\,2$

f) Differential pressure ratio $\frac{\triangle P}{P_1} = x = 0{\cdot}025\,7$

A-3.2 Computation of α

From **10.1.7.1** and **10.1.7.2**:

$$\alpha = \alpha'' \times r_{\text{Re}}$$

and $r_{\text{Re}} = (r_0 - 1)\left[\frac{\log_{10} Re_D}{5{\cdot}5}\right]^2 + 1$

For this problem $\frac{D}{k} = \frac{102{\cdot}23}{0{\cdot}04} = 2\,555{\cdot}75$ and $\beta^2 = 0{\cdot}16$

Therefore from Table 13 $r_0 = 1$, then r_{Re} is also 1 and $\alpha = \alpha''$ which can be determined as a function of β^4 and Re_D from Table 12. Since q_m is unknown Re_D is also unknown and therefore, α has to be determined by trial and error procedure.

A-3.3 Expansibility Factor

From **10.1.7.3**:

$$\epsilon = \left[(1-x)^{2/\kappa}\left(\frac{\kappa}{\kappa-1}\right)\left(\frac{1-(1-x)^{\frac{\kappa-1}{\kappa}}}{x}\right)\left(\frac{1-\beta^4}{1-\beta^4\,(1-x)^{2/\kappa}}\right)\right]^{\frac{1}{2}}$$

Substituting the values of x, κ and β

$\epsilon = 0{\cdot}9805$

A-3.4 From **8.1.1**:

$$q_m = \alpha\,\epsilon\frac{\pi}{4}\,d^2\sqrt{2\triangle P\,\rho_1}$$

$$= \alpha\,(0{\cdot}980\,5)\frac{\pi}{4}(0{\cdot}040\,89)^2\sqrt{2 \times 9\,708{\cdot}6 \times 3{\cdot}508\,3}$$

$$= 12{\cdot}875\,5 \times 10^{-4} \times \sqrt{6{\cdot}812\,1 \times 10^4}\,\alpha$$

or $q_m = 0{\cdot}336\,\alpha$ .. (Eqn C)

And From **A-1** (r)

$Re_D = 8{\cdot}896\,3 \times 10^5\ q_m$ .. (Eqn D)

TRIAL 1

Let $q_m = 0{\cdot}33$ kg/s

From Eqn (D) $Re_D = 8{\cdot}896\,3 \times 10^5 \times 0{\cdot}33 = 2{\cdot}935\,7 \times 10^5$

and $\beta^4 = 0{\cdot}025\,6$

From Table 12

$$\alpha = \alpha'' = 0{\cdot}993 + \frac{(0{\cdot}996 - 0{\cdot}993)\,(0{\cdot}025\,6 - 0{\cdot}02)}{(0{\cdot}03 - 0{\cdot}02)} = 0{\cdot}994\,68$$

From Eqn (C)

$q_m = 0{\cdot}336 \times 0{\cdot}994\,68 = 0{\cdot}334\,2 \neq 0{\cdot}33$, the assumed value.

TRIAL 2

Let $q_m = 0{\cdot}334\,2$

From Equation (*D*) $Re_D = 8{\cdot}896\,3 \times 10^5 \times 0{\cdot}334\,2 = 2{\cdot}973\,1 \times 10^5$ and $\beta^4 = 0{\cdot}025\,6$

From Table 12, it is clear that variation in the value of α'' for the above change in Re_D value is negligible and can be considered $= 0{\cdot}994\,7$.

So that from Eqn (C) $q_m = 0{\cdot}336 \times 0{\cdot}994\,7 = 0{\cdot}334\,2 =$ the assumed value.

$$q_m = 0{\cdot}334\,2 \text{ kg/s and, } q_v = \frac{0{\cdot}334\,2}{1{\cdot}057\,2} = 0{\cdot}316\,1\ \frac{\text{m}^3}{\text{s}} \text{ at STP}$$

A-3.5 In the above derivation

a) $k = 0{\cdot}04$ mm

b) $\dfrac{P_2}{P_1} = 0{\cdot}974\,2$ and is $> 0{\cdot}75$

c) $\dfrac{D}{k} = \dfrac{102{\cdot}23}{0{\cdot}04} = 2\,555{\cdot}75 > 1\,000$

d) $D = 102{\cdot}23$ mm and is between 50 and 1 000 mm

e) $\beta = 0{\cdot}4$, is between 0·32 and 0·8

f) $Re_D = 2{\cdot}973\,1 \times 10^5$ is within the tabulated values of Table 12.

Therefore, all the requirements for an ISA 1932 nozzle as specified in **10.1.7.1** *and* **10.1.7.2** *are satisfied and the above derivation is* valid.

A-4. FLOW MEASUREMENT WITH LONG RADIUS TYPE NOZZLE

A-4.1 Additional Data — Same as **A-3.1.**

A-4.2 The flow coefficient α can be derived from coefficient of discharge C by the relation $\alpha = C.E$

where E is the velocity of approach factor and is equal to $(1 - \beta^4)^{-\frac{1}{2}}$. Table 14 gives the values of C for different β and Re_D combinations. Since Re_D is a function of q_m, which is unknown, α should be found out by trial and error.

A-4.3 Expansibility Factor ϵ — Since the equation for expansibility *factor* ϵ, *is same for both ISA 1932 and long radius nozzle, from* **A-3.3.**

$$\epsilon = 0{\cdot}980\,5$$

A-4.4 From the Basic Equation (Clause 8.1.1)

$$q_m = \alpha\,(0{\cdot}980\,5)\left(\frac{\pi}{4}\right)(0{\cdot}040\,89)\sqrt{2 \times 9\,708{\cdot}6 \times 3{\cdot}508\,3}$$

$$\text{or } q_m = 0{\cdot}336\,\alpha \qquad \text{.. (Eqn C)}$$

$$\text{And } Re_D = 8{\cdot}896\,3 \times 10^5\, q_m \qquad \text{.. (Eqn D)}$$

TRIAL 1

Let $q_m = 0{\cdot}335$ kg/s. Therefore from Eqn (D) $Re_D = 2{\cdot}980\,2 \times 10^5$ from Table 14 for $\beta = 0{\cdot}4$

$C = 0{\cdot}990\,3$ (by interpolation)

$$\text{Now } E = \frac{1}{\sqrt{1-\beta^4}} = \frac{1}{\sqrt{1-0{\cdot}025\,6}} = \frac{1}{0{\cdot}987\,1}$$

or $E = 1{\cdot}013$ from **A-4.2**

$\alpha = 0{\cdot}990\,3 \times 1{\cdot}013$ or $\alpha = 1{\cdot}003\,1$

$q_m = 0{\cdot}336 \times 1{\cdot}003\,1 = 0{\cdot}337 \neq 0{\cdot}335$, the assumed value.

TRIAL 2

Let $q_m = 0{\cdot}337$ kg/s

From Eqn (D) $Re_D = 8{\cdot}896\,3 \times 10^5 \times 0{\cdot}337 = 2{\cdot}998\,0 \times 10^5$

From Table 14 for $\beta = 0{\cdot}4$

$C = 0{\cdot}990\,3$, $E = 1{\cdot}013$

Therefore, $\alpha = 0{\cdot}990\,3 \times 1{\cdot}013$ or $\alpha = 1{\cdot}003\,2$ from Eqn (C)

$q_m = 0{\cdot}336 \times 1{\cdot}003\,2 = 0{\cdot}337 =$ assumed value or $q_m = 0{\cdot}337$ kg/s and

$$q_v = \frac{0{\cdot}337}{1{\cdot}057\,2} = 0{\cdot}318\,8\ \frac{\text{m}^3}{\text{s}} \text{ at STP}$$

A-4.5 Similar to ISA 1932 Nozzle (**A-3.5**) all the requirements for the validity of the above computation are satisfied and thus the calculations from **A-4.2** to **A-4.4** are valid.

A-5. FLOW MEASUREMENT WITH VENTURI NOZZLE

A-5.1 Additional Data — Same as **A-3.1**

A-5.2 Computation of α, from **12.6.1**, $\alpha = \alpha_0 \times r_o$ for $\frac{D}{k} = 2\,555{\cdot}75$ and

$\beta^2 = 0{\cdot}16$

r_0 = can be considered as 1 (*see* Table 13). From Table 16, $\beta^4 = 0{\cdot}025\,6$,

$\alpha = 0{\cdot}995\,2$ (by linear interpolation).

A-5.3 Expansibility factor ϵ, from **12.8**:

$$\epsilon = \left[\left(\frac{\kappa\, \tau^{2/\kappa}}{\kappa - 1} \right) \left(\frac{1 - \beta^4}{1 - \beta^4 \tau^{2/\kappa}} \right) \left(\frac{1 - \tau^{\frac{\kappa - 1}{\kappa}}}{1 - \tau} \right) \right]^{\frac{1}{2}}$$

Substituting the values of τ, κ and β, $\epsilon = 0{\cdot}980\,5$.

A-5.4 Therefore from **8.1.1**

$$q_m = (0{\cdot}995\,2)\,(0{\cdot}980\,5)\,\frac{\pi}{4}\,(0{\cdot}040\,89)^2\,\sqrt{2 \times 9\,708{\cdot}6 \times 3{\cdot}508\,3}$$

or $q_m = 0{\cdot}334\,4$ kg/s and $q_v = \frac{0{\cdot}334\,4}{1{\cdot}057\,2} = 0{\cdot}316\,3\,\frac{\text{m}^3}{s}$ at STP

A-5.5 a) $Re_D = 8{\cdot}896\,3 \times 10^5 \times 0{\cdot}334\,4 = 2{\cdot}974\,9 \times 10^5$ for $\beta^4 = 0{\cdot}025\,6$. This is within the limiting values of Re_D as specified in Table 18 for venturi nozzle

b) $\beta = 0{\cdot}4$ and is between 0·32 and 0·77

c) $D = 102{\cdot}23$ mm is within the limiting values of 65 mm and 500 mm

d) $D/k = 1\,000 > 400$

e) $\frac{P_2}{P_1} = 0{\cdot}974\,2 > 0{\cdot}75$

Therefore, all the requirements are fulfilled and the above method of flow rate computation for venturi nozzle is valid.

A-6. FLOW MEASUREMENT WITH VENTURI TUBE WITH A ROUGH-CAST CONVERGENT

A-6.1 Same as **A-3.1**.

A-6.2 From **11.6.2**, $C = 0{\cdot}984$, $E = (1 - \beta^4)^{-\frac{1}{2}} \doteq 1{\cdot}013$ flow coefficient $\alpha = 0{\cdot}984 \times 1{\cdot}013 = 0{\cdot}996\,8$.

A-6.3 Expansibility factor for venturi tube is same as venturi nozzle (*see* **11.7** and **12.7**) and therefore from **A-5.3**, $\epsilon = 0{\cdot}980\,5$.

A-6.4 From **8.1.1**, mass flow rate

$$q_m = (0{\cdot}996\,8)(0{\cdot}980\,5)\,\frac{\pi}{4}\,(0{\cdot}040\,89)^2 \times \sqrt{2 \times 9\,708{\cdot}6 \times 3{\cdot}508\,3}$$

or $q_m = 0{\cdot}335$ kg/s and $q_v = \dfrac{0{\cdot}335}{1{\cdot}057\,2} = 0{\cdot}316\,9\ \dfrac{\text{m}^3}{\text{s}}$ at STP

A-6.5 a) $Re_D = 8{\cdot}896\,3 \times 10^5 \times 0{\cdot}334\,9$
$= 2{\cdot}979\,3 \times 10^5$ and is between 2×10^5 and 2×10^6

b) $\beta = 0{\cdot}4$ and is between $0{\cdot}3$ and $0{\cdot}75$

c) $D = 102{\cdot}23$ mm and is between 100 and 800 mm

d) $\dfrac{P_2}{P_1} = 0{\cdot}974\,2 > 0{\cdot}75$

Therefore, the flow rate as computed in **A-6.4** is valid.